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THE

ENGINEER AND MACHINIST'S ASSISTANT.

THE ENGINEER

AND

MACHINIST'S ASSISTANT:

A SERIES OF

PLANS, SECTIONS, AND ELEVATIONS,

OF

STATIONARY, MARINE, AND LOCOMOTIVE ENGINES,

WATER WHEELS, SPINNING MACHINES, TOOLS, ETC., ETC.,

TAKEN FROM

MACHINES OF APPROVED CONSTRUCTION.

WITH DETAILED DESCRIPTIONS,

AND

PRACTICAL ESSAYS ON VARIOUS DEPARTMENTS OF MACHINERY.

NEW AND IMPROVED EDITION.

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PUBLISHERS' PREFACE.

The object of the present Work is to supply what the projectors conceive to be a desideratum in the education of a large and important class of the community. The prominent lead which Great Britain has taken among civilized nations in the Arts and Manufactures, and in every description of constructive industry, has called into existence a numerous body of artizans and operative mechanics, many of whom are engaged exclusively in the production of those Machines, or TOOLS, in the widest sense of the term, by the agency of which the various sources of mechanical power are rendered available for facilitating and perfecting the several processes in the Arts.

Among this class of artizans, the nature of the technical operations daily coming under observation demands the exercise of more than ordinary intelligence, and the habits of mind naturally resulting from this circumstance seem peculiarly calculated at once to stimulate, and to supply the power of gratifying, the ambition of the young Engineer and Machinist to rise above the manual drudgery of his trade, and to aspire to its more honourable and lucrative situations. Accordingly, we find that those Mechanicians who have shed lustre upon their age and country by their inventive genius, or who have attained distinction by their skill in adapting and applying the inventions of others, or by their judicious management of factories in which large bodies of workmen are employed, have themselves, in the great majority of instances, risen from the ranks of operative mechanics, whose technical education it therefore becomes highly desirable to facilitate to the utmost.

Greatly as the means of popular education, and the consequent diffusion of general intelligence have been extended in recent years, the wants of this interesting portion of the community appear to have been comparatively unattended to. While in many other professions and trades Treatises from time to time were produced embodying the results of experience, and the instructions of able professors for the guidance of the youthful aspirant, no similar provision, on an adequate scale, has hitherto been made for the professional education of the practical Engineer and Mill-wright. The ordeal of a tedious apprenticeship, eked out by many years of subsequent experience, are necessary to enable him to acquire the requisite skill in his art; while his generally superior intelligence seems to render him peculiarly susceptible of that higher species of instruction which it is the object of this Work to supply.

Numerous treatises no doubt exist, in which the theoretical principles of mechanical Engineering are ably set forth, and descriptive essays on various individual machines abound, but to the practical mechanic, who has rarely the opportunity of acquiring that preliminary mathematical education necessary for the understanding of them, the former are in most instances as sealed books, while the latter, from the want of classification and reference to some leading principles, are of little value beyond the immediate purpose for which they were published. The science of constructive mechanics is, in its leading features, purely experimental, and all progress or improvement in the art depends mainly and directly upon the knowledge of what has been already achieved in it. It therefore appears to the projectors of the present Work that it will tend greatly to promote the progress of the young Engineer to lay before him an extensive collection of accurate drawings, on a large scale, of various machines of approved modern construction, exhibiting their minute detail and accurate measurements, and accompanied by descriptions, of a lucid and practical character; and, by means of Essays or Treatises on the various departments of Engineering illustrated by the Plates, to reduce the whole to a system, and make them subservient to the elucidation of the theoretical principles on which their action depends. Such a Work seems calculated not only to facilitate the *technical* education of the working mechanic, but also to serve the higher purpose of exhibiting the application, and demonstrating the value of those theoretical and mathematical investigations, without a knowledge of which he can scarcely hope to attain eminence in the higher departments of his art.

Although mainly designed for, and principally addressed to the class of operative mechanics, it is believed that such a Work as the present may also prove highly useful in conveying practical information to those whose mechanical knowledge is purely theoretical. Those who occupy themselves with investigating and reducing to system the combinations of machinery, which form such an important feature in our national industry, cannot be expected to have many opportunities of personally inspecting the workshops where that machinery is produced, or of becoming conversant, from their own observations, with the results of experience in reference to it. Such knowledge is obviously of the utmost consequence as a basis for theoretical investigation, and a Work like the ENGINEER AND MACHINIST'S ASSISTANT appears calculated to convey to such students much important practical information which had never previously extended beyond the workshop.

In consequence of the diversified and almost boundless field for illustration, the greatest difficulty experienced in forming the series of Plates which constitutes the principal feature of this Work, has been that of selection and arrangement. The constant aim has been to furnish a series of examples which should illustrate some generally useful principles in mechanics, and, at the same time, to present such a variety of machines as should render the work most extensively acceptable as a book of reference to those engaged in departments of constructive mechanics widely differing from each other. Accordingly, the Illustrations will be found to be of a somewhat miscellaneous character; it is believed, however, that none of the more important departments of Engineering operations has been left wholly unillustrated, while some of them, such as Tools and Water Wheels, have been very fully treated. The specimens selected are from the best examples at present in operation, and in the execution of the

Plates the utmost regard has been had to that strict accuracy which the nature of the subject imperatively demands. The scales to which the Engravings are laid down are, in most instances, sufficiently large to render them available as working drawings for the reproduction of similar machines; and, where this was not practicable on account of the great dimensions or intricacy of the parts, details on a larger scale have been added to the general views. It was originally intended to make the work combine instructions in mechanical drawing with its other objects, and, accordingly, a few of the earlier Plates and descriptions are constructed after this plan. Although most of the Engravings are admirably adapted for *copies* for the student of mechanical drawing, the idea of giving instructions in this art was speedily abandoned, as it was found to complicate the work, and to obstruct its other more special objects.*

The Descriptions of the Plates are treated with the same regard to minuteness and accuracy which it has been the aim of the Publishers to introduce into the Plates themselves. Instead of a bare enumeration and detail of the parts, the descriptions will be found, in most instances, to approach more to the character of short essays on the Machines to which they are severally devoted; embodying brief historical notices, statements as to the general design and economy of the machines, with other collateral information, descriptions of their action, and such remarks and comparisons upon the different modes of construction as the practical experience of the writers enabled them to make. The style of language employed in these descriptions is such as, while familiar to the technical, will not, it is believed, prove unintelligible to the general reader who may be desirous of information on such subjects.

The Essays, which form the principal portion of the letter-press, are each devoted to the consideration of one special and important branch of Engineering practice, and are designed to afford the means at once of classifying and arranging under certain distinctive heads the subjects illustrated by the Plates, of recording the history, progress and present condition of those departments of the art to which they severally refer, and of showing the application of the theoretical principles of mechanics to practice This portion of the Work consists of the following original papers:—

I. A Practical Treatise on the Construction and Application of the Steam Engine, from the pen of a gentleman who has enjoyed peculiarly favourable opportunities for observing and recording the most recent and valuable improvements which have been made in that department of mechanical operations.

II. A Treatise on Mill Geering, in which the author has elaborately collected and systematized the most important facts and doctrines connected with a department of the engineer and millwright's practice which has engaged the attention of many of the most eminent mathematicians both of this country and of the Continent.

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* The Publishers are at present engaged in the preparation of a Manual of Instructions on Mechanical Drawing, with Specimens illustrating the progressive stages in the student's course, from the simplest elements to the most complicated combinations of machinery, and they feel confident that, from their increased experience and extended sources of information, they will be enabled to produce such a Work as will prove highly acceptable and useful.

III. An Essay on Tools, devoted to the consideration of those machines which are employed in our workshops to facilitate the production of other pieces of machinery. In this Essay, which is emphatically of a practical character, the examples illustrated by the Plates are classified, the leading principles involved in their construction explained, and such collateral information given as could not, with propriety, be introduced into the descriptions of the Plates themselves.

IV. A Dissertation on Water Wheels, containing many theoretical investigations and much practical information on this subject which has not hitherto been submitted to the public.

In addition to the large Plates, the Work is copiously illustrated by Woodcuts interspersed throughout the text. While no expense has been spared by the Publishers in securing the highest degree of excellence both in the illustrated and letter-press departments, by the arrangements they have adopted they will be enabled to issue it at such a moderate cost as will render it attainable by the humblest artizan.

The Publishers present their grateful acknowledgments to those gentlemen who have kindly furnished drawings and information regarding the various machines illustrated in this Work. In the alacrity uniformly manifested to supply the necessary materials, they are gratified to recognize an appreciation of the value of their labours on the part of those best qualified to judge of them. To their numerous correspondents who have from time to time addressed them on subjects connected with the Work, they would also return thanks, not only for the pleasure they have felt in this mark of the general interest taken in the progress of their labours, but also for many valuable suggestions, of which they have frequently availed themselves.

Glasgow, *July* 1850.

ANALYTICAL

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ENGINEER AND MACHINIST'S ASSISTANT.

PART FIRST.

THE CONSTRUCTION AND APPLICATION

OF THE

STEAM ENGINE.

BEFORE entering upon the main business of this essay, which professes to treat of the construction and working of the Steam Engine of the present day, it will be necessary to devote a short introductory chapter to an exposition of the great natural laws which are concerned in the operations hereafter detailed. This will be followed by a brief historical sketch of the invention.

SECTION I.

EXPOSITION OF NATURAL LAWS AFFECTING THE STEAM ENGINE.

Pressure of the Atmosphere.—It was discovered by Torricelli, early in the 17th century, that the air was a material substance, possessed in common with all other fluids of weight, elasticity, and compressibility. The weight of the atmospheric column varies with the height at which we stand above the level of the sea, and also at the same place, at different times, as indicated by the *barometer*. This instrument is formed by filling a tube, of about thirty-four inches in length, with mercury, one end being sealed, and the other open, when, on reversing the tube, and dipping the open end in a cup of mercury, the mercury in the tube will be found to subside to the height of about thirty inches above the surface of that in the cup—a portion of the upper part of the tube being left empty, or *in vacuo*.

We thus infer that the weight of the atmosphere, near the level of the sea, balances a column of about thirty inches of mercury, which is equal to a pressure of about 15 lbs. on every square inch of surface.

If we construct a barometer with water in place of mercury, we shall find the column subside in the tube at the height of about thirty-three feet, proving that the weight of the atmosphere balances a column of water of this height. From the mobile nature of fluids, the pressure is, of course, the same in all directions, whether upward, downward, or lateral.

Elasticity of Fluids.—All bodies are more or less compressible, and all tend to regain their original form when the pressure is removed, so that all bodies are elastic, although in very different degrees. Air, steam, and the other gases, possess these properties in a very high degree; and it is remarkable that the law of the diminution of their bulk, from pressure, is the same for all, and, within certain limits, very regular; their bulk or volume varying simply in the inverse ratio of the pressure, so long as the temperature remains a constant quantity. Thus, with a two-fold pressure their bulk is diminished to a half, with a three-fold pressure to a third, and so on; but this law does not hold good in extreme cases, either of pressure or rarefaction, becoming more disturbed according as the gas approaches the point at which liquefaction from pressure may be expected. Water and other liquids are also compressible, although to a very small extent. The amount assigned by Professor Oersted for the compression of water is but 47 millionth parts of its bulk for one atmosphere, (or $\frac{1}{212}$th part for 100 atmospheres), alcohol 22 millionth parts, and mercury only 2·6. Water will regain its volume, even from this slight degree of compression, with considerable violence, which shows that it is not deficient in elasticity.

Expansion from Heat.—It may be considered, as a fundamental principle, that heat causes expansion in all bodies, although there are a few remarkable exceptions. This expansion is never a permanent effect, but lasts only so long as the exciting cause is present. Its amount varies exceedingly in different bodies, being greatest in the gases, next in liquids, and least in solids. The gases also differ from liquids and solids, by all expanding to an exactly equal degree by the same increase of temperature, as well as in the law of their expansion being more uniform—that is to say, they maintain the same degree of expansion for each successive and equal increment of temperature.

The Thermometer.—Owing to these properties of the gases, the expansion of air was the first means employed for measuring heat by means of the air thermometer, which is believed to have been invented by Galileo and Sanctorio, in the beginning of the 17th century; but as the indications of such an instrument are necessarily uncertain, owing to their dependence upon atmospheric pressure, they were soon disused, and alcohol was the next substance employed for this purpose, by the Florentine academicians. Dr. Halley first suggested the boiling point of a liquid as a fixed point from which to graduate a scale, and as the temperature of the boiling point is dependent upon the pressure of the atmosphere, he made the caves under Paris his standard of elevation. Amonton next adopted the boiling point of water as a standard, but Sir Isaac Newton was the first who took *two* fixed points—those of freezing and boiling water, as the constant points between which to graduate his scale. The liquid he employed for his thermometer was linseed oil, which he preferred to alcohol, on account of its higher boiling point; but both these liquids are objectionable, owing to the uncertainty which must exist as to their state of purity. Newton divided his scale between the freezing and boiling points of water, into thirty-four degrees. Fahrenheit, a Dutchman, was the first who introduced mercury as the expanding fluid in thermometers, for which it possesses many advantages. These are, the great range it affords between its boiling and freezing points, the degree of purity, and hence, of uniformity, in which it is obtained by distillation, and its uniform expansion for equal increments of temperature, at least within moderate limits. The law of expansion in all fluids becomes more disturbed as they approach their freezing or boiling point, being then either prematurely increased or diminished, as if in anticipation of the great change which is about to take place amongst their particles. In this point of view, mercury ranks before all other fluids, but it yields to alcohol in expansibility, as well as in the lowness of its freezing point. Fahrenheit divided his scale between freezing and boiling water into 180°, fixing his zero at 32° below the freezing point of water. He seems to have adopted his zero point from the very erroneous belief, that the cold produced by a mixture of snow and salt was the greatest that could be obtained by artificial means. In his thermometer, one ten-thousandth part of the capacity of the bulb indicates one degree. Reaumur's modern thermometer has the scale divided into 80° between the freezing and boiling points of water; and that of Celsius, commonly called the Centigrade scale, is divided into 100°, between the boiling and freezing points. Hence, the lengths of a degree respectively, in the scales of Fahrenheit, Centigrade, and Reaumur, are as the numbers 180, 100, 80, or as 9. 5. 4. In old thermometers it is invariably found that the freezing point has risen a little, owing, probably, to a slight contraction in the glass bulb. Alcohol is still employed for thermometers in cases where intense cold is required to be indicated, this being the only liquid which has never been congealed by cold, although its temperature has been reduced, by artificial means, to—162° Fahrenheit, or 194° below the freezing point of water. Mercury, on the other hand, freezes at —40°, and in the act of congelation, suddenly shrinks in volume to a considerable extent. The following substances expand cubically for 100° of Fahrenheit: air, $\frac{1}{2\cdot66}$ part of its bulk, (or $\frac{1}{480}$ for 1°); mercury, $\frac{1}{55\cdot5}$; glass, $\frac{1}{387}$; iron, $\frac{1}{282}$; copper, $\frac{1}{195}$; platinum, $\frac{1}{377}$.

Pyrometers.—Various descriptions of pyrometers have been invented for measuring greater degrees of heat than can be shown by the thermometer. The best is that of Professor Daniell, which is thus constructed. A case of black lead is first provided, that substance being very slightly expansible and not easily cracked by the heat. Within this case, which is fitted with a cover of the same material, there is loosely placed a bar of platinum. A plug of porcelain is made to slide tightly through a hole in the centre of the cover, and when the apparatus is cold, this rests upon the head of the platinum bar; but when placed in the furnace, the linear expansion of the platinum pushes the porcelain plug through the hole in the cover, and, as it was tightly fitted, it preserves the position to which it has been protruded after the platinum bar has contracted by removal from the fire. Wedgewood's pyrometer depends for its action on the anomalous contraction of clay by heat, but it cannot be depended upon in its results. From the unsatisfactory nature of the experiments made with every kind of pyrometer hitherto constructed, we are left in great uncertainty as to the actual amount of caloric present in the production of very high temperatures. This is seen by the contradictory results of different experiments made to ascertain the fusing points of the more refractory metals. Thus, three of the chief experimenters on this subject give the following as the result of their researches, viz.:—

	By Daniell.	By Wedgewood.	By Morveau.
Visible red heat at	980° Fahr.	1077° Fahr.	517° Fahr.
Copper melts at	2548 —	4587 —	2205 —
Silver melts at	2233 —	4717 —	1823 —
Gold melts at	2590 —	5237 —	2518 —
Cast-iron melts at	3479	about 20,000 —	8696 —

Of these, Daniell's are probably the most correct.

Anomaly in the Expansion of Water.—We have said that all liquids expand by heat. Water, however, presents a remarkable anomaly in its law of expansion, since it *contracts* by heat from its freezing point to the 40th degree of Fahr., at which it attains its maximum density, and expands from thence regularly to the boiling point. This remarkable property of water is distinctly shown by an ingenious apparatus invented by Sir John Leslie, in which a freezing mixture is applied to the centre of a column of water, and thermometers are inserted at the top and bottom to indicate the change of temperature as the density of the water varies. The effect of the freezing mixture (which, it must be remembered, is applied in the *middle* of the column of water), is first to cause a descending current of cold water, until the temperature of 40° is attained by the whole below the freezing mixture; after this, an ascending current of cold water takes place, and ice begins to form on the surface. It must strike every one what a beautiful provision of nature this is for man's comfort and enjoyment, that in the case of water alone, of all other fluids, the ordinary law of contraction should be dispensed with; for had it been the case that water increased in density in proportion as its temperature fell, we should have had our lakes and rivers freezing from the bottom, and that so rapidly, that in a short time the whole would have become a solid mass of ice which the summer heat could never thoroughly melt, owing to the low conducting power of water. The true point of maximum density of water, as ascertained by a great number of experiments, occurs at 39.23° Fahr. In brine, this point is a little lower.

Expansion of Solids and Liquids.—Of solids, by far the most expansible are the metals; and in speaking of their expansion we generally imply the *linear* expansion of a rod of the metal, and not the cubical expansion as for liquids or gases. The following is the amount of expansion for different solids, in a fraction of their length, between 32° and 212° Fahr., viz.:—

SOLIDS.	
Glass and Platinum,	$\frac{1}{1130}$
Cast-iron and Steel,	$\frac{1}{900}$
Wrought-iron,	$\frac{1}{850}$
Copper and Brass,	$\frac{1}{580}$
Gold,	$\frac{1}{650}$
Silver,	$\frac{1}{520}$
Tin,	$\frac{1}{420}$
Lead,	$\frac{1}{350}$
Zinc,	$\frac{1}{300}$
LIQUIDS.	
Mercury (cubical expansion),	$\frac{1}{55}$
Water (do., from point of maximum density),	$\frac{1}{23}$
Water (do., saturated with common salt),	$\frac{1}{20}$
Alcohol, (do., to its boiling point),	$\frac{1}{17}$

Some solids deviate from the general law of expansion by heat, as the alloy of bismuth, lead, and tin, called fusible metal, which melts at 201° Fahr., its point of maximum density being at 156°, and its minimum at 111°.

Liquefaction and Congelation.—As a general principle, we may consider that the molicular change of liquefaction is induced in all bodies by heat. Every substance with which we are acquainted has been liquefied by its influence, except carbon; the most refractory metals, and even the pure earths, having yielded to the intense heat of the oxyhydrogen blowpipe, or the galvanic current. In most metallic alloys the fusing point is different from that of either of the metals singly; and also differs from the mean fusing points of its constituents—thus, the alloy of two bismuth, one tin, one lead, melts in water of 200° temperature, although neither of these metals, separately, melts at a temperature below 600°. The freezing, or fusing, point of mercury (for both terms signify the same thing), was not discovered till 1759, at St. Petersburg, although it freezes by natural cold in many parts of Russia and Siberia, Nova Zembla, &c. The greatest natural cold on record (shown by the spirit thermometer), is —70° Fahr., or 30° below the freezing point of mercury. Probably all fluids have been frozen except pure alcohol, which has been subjected to the greatest artificial cold hitherto produced, viz., —162° Fahr., without showing any symptoms of congelation. Brandy freezes at —57°, and sulphuric æther becomes viscid at —50°. Mercury, unlike water, contracts much in freezing, whilst cast-iron, bismuth, and antimony, expand in solidifying. Liquid carbonic acid freezes at —160° Fahr., this extreme cold being produced by the expansion of the liquefied gas when relieved from pressure. The experiment is performed with a strong iron bottle, provided with a stop-cock, the carbonic acid gas being generated as well as compressed within the bottle itself, where it assumes the liquid form when the pressure exceeds thirty-six atmospheres, at 32° Fahr. If the stop-cock be now opened, and the liquid allowed suddenly to expand into gas and escape through the neck of the bottle, such an intense cold is produced, that a part of the gaseous carbonic acid is first changed back again into the liquid form, and then frozen at the neck of the bottle, where it collects in the form of snow. This singular effect is produced, as will presently be more fully explained, by the amount of "sensible" heat which is withdrawn from the surrounding air, to supply the requisite quantity of "latent" heat for the gas in its expanded form. The cold thus produced will instantly freeze mercury, yet the solid carbonic acid has such a low conducting power for heat, that a portion of it may be handled, or even placed on the tongue, with impunity, whilst a globule of frozen mercury, although not nearly so cold in itself, will blister the skin like molten lead.

Latent Heat of Water.—All bodies have more or less heat in combination with them which is not sensible to the thermometer—hence called "latent heat." This doctrine was first expounded by Dr. Black of Glasgow, in 1762. In water, especially, a large quantity of latent heat is present. This fact is made apparent in the melting of ice at 32° into water at 32°, all the heat which has been supplied for this purpose being rendered latent in the water, the thermometer taking no account of it. We find by experiment that it requires as much heat to change ice at 32° into water at 32°, as if we had added 140° to the solid ice, were it capable of receiving this temperature without undergoing the molicular change of liquefaction; or, what is the same thing, as much heat as would raise ice-cold water of the same weight to 172°. When water is again changed into ice, both its latent and sensible heat must first be withdrawn by the surrounding atmosphere, which accounts for the slowness of the process of congelation in ordinary circumstances. All bodies absorb more or less latent heat during liquefaction; tin absorbs 500°. It is from this general principle that freezing mixtures derive their energy, a rapid and artificial liquefaction being induced in the ice or snow employed by means of the chemical action of the salts—the effect of this forced liquefaction being to absorb rapidly a large quantity of sensible heat from all objects in the vicinity, in order to supply the latent heat demanded by the water in its liquid form. The greatest amount of cold that has been produced in this way is —90° Fah. The converse of this action is when water, from any cause, becomes suddenly solid, when a portion of its previously latent heat becomes sensible, as might be expected.

This is well shown by permitting a saturated solution of Glauber's salts to cool undisturbed in vacuo (by corking and removing the flask while the liquid is boiling), when, on the re-admission of air, the whole suddenly crystallizes, and its temperature rises at the same moment some forty or fifty degrees. Water in a perfectly still atmosphere, and under favourable circumstances, has been reduced to a temperature of 5° Fahr. without freezing; but in such a case, upon the slightest agitation, the whole water becomes instantly solid, and its temperature changes at once to 32°, owing to a portion of its latent heat being liberated and thus becoming sensible to the thermometer.

Latent Heat of Steam.—We have seen that water, in passing from the solid to the liquid form, absorbs a quantity of latent heat, and we find that while passing into the gaseous form it absorbs a large quantity more, the actual amount varying with the temperature of the water which is changed into vapour. This quantity, according to the recent experiments of Regnault, is 965° for water boiling at 212°—that is to say, that if the water were capable of having its temperature raised above its boiling point without undergoing the change of vaporization, such water would be raised to a temperature of 1177° by the amount of caloric which has been expended in changing it into steam of 212°. Hence we say that the latent heat of steam of the atmospheric pressure, (or 212°), is 965°; but this quantity varies in a constant proportion with the temperature of the vapour, the sum of the latent and sensible heat of steam or vapour of any density or temperature being always the same number, viz., 1177°. Thus, steam of 112° has 1065° of latent heat; steam of 212° has 965° latent; steam of 312° has 865° latent heat, &c. Hence we find that the operation of boiling in vacuo (although of much practical utility in some of the arts), effects no saving in the consumption of fuel, the same amount of fuel being required for raising steam of any pressure or temperature, taking, of course, equal *weights* of steam at a given elasticity.

The following table contains the latent heats of vapour and steam of different temperatures, according to the recent, and, it is believed, very accurate, experiments of M. Regnault.

Fahrenheit.		Fahrenheit.	
Sensible Heat.	Latent Heat.	Sensible Heat.	Latent Heat.
32·000	1091·7	249·450	940·1
50·003	1079·1	267·794	927·2
68 018	1066·7	286·187	914·4
86·046	1054·2	304·632	901·2
104·091	1041·6	323·133	888·5
122·156	1028·9	341·693	875·1
140·246	1016·4	360·316	862·2
158·381	1003·7	379·002	848·9
176·507	991·1	397·760	835·7
194·685	978·3	416·588	822·2
212·900	965·7	435·480	808·9
231·153	952·9	454·474	795·4

Boiling Point of Liquids varies with the Pressure.—The boiling point of liquids is wholly dependent upon the pressure under which this operation proceeds. Thus, we assume our standard point at 212°, because water boils at that temperature when the pressure of our atmosphere equals 30 in. of mercury. In a tolerably perfect vacuum under the air pump, water boils at a temperature of 100°, but in a perfect vacuum it will boil at 72°, according to Professor Robison, who came to the conclusion that all fluids boil in vacuo at about 140° below their ordinary boiling point. On the other hand, the boiling point of liquids rises in temperature according as the pressure is increased, so that we have an invariable connection between the temperature of the boiling water with its resulting steam, and the pressure under which ebullition takes place—or, in other words, between the temperature and the pressure of *steam*. To determine this relation with accuracy, has been the object of extensive experiments in all countries, and we subjoin the following table of the results for high-pressure steam, obtained by the French Academy and the Franklin Institute, reserving for another place a more extended table of the temperatures and volumes of steam at the more ordinary densities.

Pressure in atmospheres.	Temperature. Fahrenheit.		Pressure in atmospheres.	Temperature. Fahrenheit.		Pressure in atmospheres.	Tempera. Fahr.
	French Academy.	Franklin Institute.		French Academy.	Franklin Institute.		French Academy.
1	212°	212°	6	320·4	315·5	12	374
1½	233·9	235	6½	326·3	321	13	380·7
2	250·5	250	7	331·7	326	14	386·9
2½	263·8	264	7½	336·9	331	15	392·9
3	275·2	275	8	342	336	16	398·5
3½	285·1	284	8½		340·5	17	403·8
4	293·7	291·5	9	350·8	345	18	408·9
4½	301·3	298·5	9½		349	19	414
5	308·8	304·5	10	358·9	352·5	20	418·5
5½	314·2	310	11	366·8		21	423

It should be observed that the atmosphere of the French Academy corresponds to the pressure of ·76 of a French mètre of mercury, or 29·921 English inches.

Advantage has been taken of the connection between the boiling point of water and the pressure of the atmosphere, to construct a very portable and sensitive barometer, by which the atmospheric pressure is determined with the utmost nicety, from observing the boiling point of water in a little metallic vessel by the aid of a peculiarly delicate thermometer, whose scale need not extend beyond a few degrees above and below the ordinary boiling point. Dr. Francis Wollaston, its inventor, used to assert that he could give this instrument such a delicacy as to indicate the difference of height, not only between the top and bottom stories of a house, but even between the ceiling and the floor of a room. It is apparent that it requires the same correction for the variable pressure of the atmosphere, as the common barometer does, when employed for measuring heights. At the top of Mont Blanc, water boils at 186° Fahr.

Boiling Point of Sea Water.—We have hitherto considered the temperature and pressure of steam raised from pure water only, but when sea water is employed we obtain very different results, according as it is more or less saturated with salt. From the most careful analysis, sea water has been found to contain the following salts in solution, viz., chloride of sodium (common salt), 2·50; chloride of magnesium, 0·35; sulphate of magnesia, 0·58; carbonates of lime and magnesia, 0·02; sulphate of lime, 0·01; water, 96·54, in 100 parts. It also contains iodide of sodium and bromide of magnesium in small quantities. Its average specific gravity is from 1·029 to 1·030, at which it boils at the temperature of 213·2°. As the salt does not evaporate with the steam, the water, of course, gradually increases in saturation and density; its boiling point, and consequently the temperature of the steam, continuing to rise in proportion. The following table gives the temperature of steam at the atmospheric pressure, (or of the boiling point), for sea water at different stages of saturation.

Proportion of salt in 100 parts, by weight.		Temp. Fahren.	Proportion of salt in 100.		Temp. Fahren.
Common Water,	—	212°	Sea Water,- -	21·22	220·2
Sea Water,- -	3·03	213·2	... - -	24·25	221·4
... - -	6·06	214·4	... - -	27·28	222·5
... - -	9·09	215·5	... - -	30·30	223·7
... - -	12·12	216·7	... - -	33·34	224·9
... - -	15·15	217·9	Saturated solution,	36·37	226·0
... - -	18·18	219·0			

Volume of Vapours.—The volume of steam at the boiling point into which one measure of water expands is 1711, or in rough numbers, one cubic inch of water becomes a cubic foot of steam at 212°. The same number for alcohol is 476, and for sulphuric æther 220; one volume of water producing more vapour than an equal quantity of any other known liquid.

Condensation.—Steam may be condensed by any body colder than itself, but water is usually employed in the steam engine for this purpose, as it has a great capacity for heat, and may be made to present a very large cooling surface to the steam. The quantity of water thus employed depends upon the temperature of the steam in the cylinder, and will be given under the head of the condenser.

Evaporation.—We have seen that it is a law of universal application, that a liquid cannot undergo any change in its condition without the addition or subtraction of heat. Thus, for every pint of water which rises by spontaneous evaporation from the surface of the earth, just so much heat has been consumed in effecting the change, as if the same quantity of water had been boiled away over a fire; indeed, more heat is required, as the cold water must absorb so much heat as would first have raised it to the boiling point, in addition to the 965° of latent heat proper to the steam. This sufficiently explains the cooling effect of evaporation, which, in the more volatile fluids, is very intense, especially if assisted by a diminution of pressure. Æther, whose boiling point, under the pressure of the atmosphere, is 96°, boils under a pressure of nineteen inches of mercury at 60°, producing by its evaporation so intense a cold as to freeze mercury placed near it. The evaporation of water placed in a porous earthenware tray, under the receiver of the air-pump, causes such a reduction of temperature as speedily to freeze its own surface, although in a room at 60°. Dalton first refuted the idea that air held watery vapour in chemical solution; the fact being that space or vacuum, instead of air, takes up and becomes saturated with moisture, which then forms an atmosphere in its own person, and is endowed with pressure and elasticity like other gaseous fluids. Of course the atmosphere plays an important part in the process of ordinary evaporation, by supplying the necessary heat.

The following table gives the elasticities of watery vapour of very low tension, in inches of mercury.

Temperature.	Elasticity.
32°	·200 inches.
40	·262 ...
50	·375 ...
60	·524 ...
70	·721 ...
80	1·000 ...

The following table shows the specific gravities of some vapours: first, as compared with air of the same temperature; and secondly, as compared with air of 60°.

	Fahrenheit.	Specific gra.	Specific gra.
Water,............boils at	212°	0·6235	0·481
Alcohol,.......... "	173	1·6020	1·311
Sulphuric Æther, "	96	2·5860	2·415
Turpentine,....... "	314	5·0130	3·342

Specific Heat.—The "specific heat" of a body, by which is meant its capacity for latent heat, is quite independent of the density of the body. Thus, the specific heat of air is only ¼ that of water, for equal weights, although it is 800 times rarer, and the specific heat of water is 30½ times that of mercury—that is to say, it will take 30½ times more heat to raise a pound of water to 212° than a pound of mercury. Even taking equal bulks of these two fluids, the water will require 2¼ times the heat that the mercury will. The specific heats of the following bodies, as compared with water, have been obtained by careful experiment, the method employed being to observe the weight of ice liquified by each when enclosed in ice, during its passage from a given temperature to the freezing point, the weight of the body forming an element in the calculation.

SPECIFIC HEATS FOR EQUAL WEIGHTS, WATER BEING UNITY.

Water,.........	1·0	Copper,........	·095	Mercury,.......	·033
Sulphur,.......	·188	Zinc,...........	·093	Gold,............	·030
Iron,...........	·110	Silver,..........	·056	Lead,......... Bismuth,....	·029

The specific heat of the gases is still very indeterminate, but it is always increased as they are rarified, and diminished as they are compressed, which it is believed is a general law for all bodies. When air is suddenly compressed to half its bulk, it rises in temperature by 65° Fahrenheit, as exemplified in pumping air; and if violently compressed by a syringe it will readily inflame tinder, a temperature of from 800° to 1000° being the result. This is, of course, owing to the latent heat in combination with the air, at its natural density, having been suddenly obliged to quit and become sensible, from the *incapacity* of the condensed air to hold it any longer in combination. Exactly the reverse action takes place when vapour or air of any density is suddenly rarified, as exemplified in the low temperature of a jet of high pressure steam; sensible heat in this case becoming latent, in order to supply the increased *capacity* of the rarified steam. This may be assigned as one of the reasons of the intense cold in the higher regions of the atmosphere, the dense currents of air from the surface of the earth absorbing heat as they become rarified by the diminished pressure.

Combustion.—All solid bodies, when heated to about 800° Fahrenheit, become luminous in the dark. The gases have never been rendered luminous, although so hot as to bring iron to a red heat. Combustion is purely a chemical operation, depending upon the rapid combination of oxygen with the combustible body, and it appears to be in all cases more or less intense, in proportion as the body is more or less entirely consumed during the process; the incombustible portions, whether solid or gaseous, tending only to abstract heat from the burning mass. Thus, in the oxyhydrogen blow-pipe, where the combustible body, the hydrogen, becomes completely consumed by combining with exactly the proper amount of oxygen (supplied in the most efficient manner) we have the most intense of all the sources of heat, except that derived from electricity. For the production of flame, a most intense heat is requisite. This fact is proved negatively by Davy's safety lamp, where the passage of the flame through the lantern is completely stopped by the intervention of thin wire gauze; the inflammable gas, although exploding within the lantern, being cooled so much during its passage through the meshes of the gauze as to be incapable of retaining its condition of flame at the other side.

Considerable heat is produced by percussion and friction, owing, probably, to the condensation of the particles of the body, which, in its condensed form, possesses a diminished capacity for heat, so that a portion is necessarily rendered sensible.

Communication of Heat.—Heat may be communicated from one body to another in three ways, either by radiation, conduction, or convection.

By Radiation.—Radiation takes place from the surface of a hot body to a colder one in its neighbourhood, so long as any difference of temperature exists between them. The Newtonian law of radiation was, that the velocity of cooling was proportional, simply, to the excess of temperature, which, for small differences of temperature, is a sufficiently near approximation; but in considering extreme differences, the Newtonian law becomes grossly incorrect, as the rate of cooling then varies in a geometrical progression with the increase of temperature of the radiating body, and is, hence, expressed by a logarithmic curve. This law was discovered by Dulong and Petit. Radiation is modified, first, by distance. Its intensity diminishes as the square of the distance, following the same law with light, magnetism, electricity, and all other influences which proceed from a centre. It is modified, secondly, by the nature of the surface of the body emitting heat, or the *radiant* body; and, thirdly, by the nature of the surface of the body receiving heat, or the *sentient* body.

The comparative roughness or smoothness of a surface has the most material influence upon the radiation of heat, which takes place with much greater facility from a rough surface than from a smooth or polished one. The following is the radiating or "emissive" power of some different bodies, lamp-black, owing to its deficiency of polish, being taken as the standard of comparison, or of greatest radiating power, viz.:—

Lamp-black,....................	100	China Ink,......................	88
Rough paper (white or black),	98	Isinglass,........................	80
Glass,..............................	90	Polished Metals, ...	12

These numbers are only applicable in the case of *non-luminous* heat, such as that from water or steam, where colour does not affect radiation; but when the heat becomes luminous, the colour of the radiant or sentient surface is found to have an important influence. Thus, if we take two thermometers, the one with a clear or white bulb, and the other with a black one, and expose them equally to the sun's rays, the clear-bulbed thermometer will not rise above 80° at the utmost, whilst the blackened thermometer may be raised above the boiling point of water. The peasants of Switzerland have long possessed a practical knowledge of the superior absorbent properties of a black over a white surface, as they are in the habit of strewing soot or mud over the surface of the snow, to assist its melting in the spring, and thus clear their pastures. We need hardly say that the same body must possess an equal facility for absorbing as for radiating heat.

By Conduction.—The curve formed by the conduction of heat along a metal bar is the logarithmic, in which the ordinates increase in a geometrical ratio. Thus, supposing we have a series of thermometers ranged at equal distances along a metal bar, which is heated at one end, if the first thermometer reach 150°, the second will be 100°, the third 75°, the fourth 63°, the fifth 57°, the sixth 54°, and so on. The relative conducting powers of the metals occur in the following order—viz., gold, silver, copper, brass, iron, zinc, platinum, tin, lead, antimony, bismuth. The worst conductors of heat are earthy and animal substances; such as clay, ashes, hair, and wool, &c. If the conducting power of gold be called 1000, that of clay will be about 12. A remarkable instance of the slow conducting power of *ashes* occurs on the flanks of Mount Etna, where an ancient glacier, which had been covered many hundred years ago by a shower of volcanic ashes, and subsequently by a lava-stream, is at the present day quarried for ice in summer, when the snow fails on the mountain. It was long doubted whether liquids were capable of conducting heat in the proper sense of the term, but it has now been proved that they are so, although in a very slight degree.

By Convection.—The ordinary method by which heat is communicated to liquids is by what is called *convection;* as the particles of liquid at the bottom of the vessel which first receive the heat are *carried up* by their diminished specific gravity, until the heat they have acquired is absorbed by the colder liquid above; their place at the bottom of the vessel being in the meantime supplied by the descent of colder and heavier fluid. By this process there is kept up a constant current of cold water descending to the bottom, and an ascending current of heated water, until the whole liquid attains the uniform temperature of its boiling point; after which ebullition commences, and the heat which now enters the water becomes latent in the steam.

As the process of conduction through liquids is very slow, it is quite possible to have water at the same time boiling at the top and freezing at the bottom of the same vessel.

Gaseous Contents of Water.—All water contains, entangled as it were amongst its particles, a considerable quantity of air and carbonic acid gas; which become partially separated by boiling, but more completely so by the joint action of heat and the air-pump. Dr. Ure's experiments, made by simply boiling the water, give the following proportions of gaseous matter in water of different kinds at 55°, viz.—

River water swollen by winter rains, ...	2.80	per cent.
Canal water in winter,.........................	2.67	—
Filtered river water,............................	2.51	—

As it cannot be supposed, however, that the whole air was extracted by boiling, we may assume that river or canal water contains nearly 5 per cent., or $\frac{1}{20}$ of its bulk of gaseous matter; and spring or well water $\frac{1}{15}$. This volume is assumed for air at the temperature of 60°, under the pressure of the atmosphere, so that when proper allowance has been made for the expansion of the gases due to the temperature of the condenser of a steam engine (about 120°), and the diminished pressure (say, two inches of mercury), it will be found that the volume has increased about fifteen times.

Air-Pump of the Steam Engine.—In considering the work to be performed by the air-pump of a steam engine, an allowance must also be made for the fixed gases which pass over with the steam from the boiler into the condenser, as well as for leakage of air through the joints, and to the whole there must be added an equal volume of uncondensed watery vapour. This calculation will show about three times the bulk of the water from injection and condensed steam, as the volume of the uncondensed gases to be withdrawn by the air-pump. Salt water contains less air than fresh, but this advantage is counterbalanced by the increased volume which the enclosed air occupies in the boiler, owing to its higher degree of temperature at a given elasticity.

SECTION II.

ON COMBUSTION AND THE PROPERTIES OF FUEL.

Before quitting this introductory part of our subject, it will be necessary to make the reader acquainted, in as concise a manner as possible, with the constituents and properties of fuel, and the laws which govern its combustion in furnaces; as it is only by paying strict attention to the undeviating principles of nature, that the practical engineer can expect to avoid occasional failures in this important branch of his profession. Our remarks and data will be drawn chiefly from the works of Turner, Reid, Dr. Ure, Mr. Parkes, and Mr. Wicksteed, the Mining Review, and the recent Parliamentary Report on Coals for the Royal Navy, by Sir Henry de la Beche, and Dr. Lyon Playfair.

Analyses of Fuel.—The following table gives the mean composition of average samples of coals, according to the Parliamentary Report:—

PROPERTIES.		Welsh Coals.			Scotch Coals			English coals, mean of two samples.	Patent fuels, mean of three samples.
		Mean of nineteen samples.	Highest.	Lowest.	Mean of five samples.	Highest.	Lowest.		
Specific gravity,........		1·31	1·39	1·25	1·26	1·316	1·20	1·266	1·13
In 100 parts.	Carbon,..........	83·4	91·44	73·84	77·4	79 85	74·55	77·61	85·9
	Hydrogen,........	4·8	5·84	3·46	5·3	5·5	5·14	5·93	5·5
	Nitrogen,........	1·2	2·16	Trace.	0·8	1·41	Trace.	1·94	0 8
	Sulphur,..........	1·5	5·07	—	1·02	1·53	0·33	2·56	1·2
	Oxygen,..........	3·65	8·29	0·39	10·36	15·51	5·05	5·42	2·5
	Ash,.............	5·54	14·72	1·50	5·1	10·70	3·10	7·03	4·1
Per centage of coke from each,........		75·5	92·9	56·	54·1	58·45	49·8	58·5	74·2

Besides the substances enumerated above, there is always present in coal a small portion of water, varying, in tolerably dry specimens, from 1 to 3 per cent. of the weight.

Wood, on the contrary, contains about ⅕ of its own weight of hygroscopic water; and the heat necessary for evaporating this large quantity exceeds 20 per cent. of the total heat obtained by the combustion of the wood, which is the principal cause of the great inferiority of this substance to coal for generating steam.

Combustion considered Chemically.—In the following table we have the different gases which are brought into action, either as inflammables, supporters, or products of combustion. The first column of figures represents the weight of an equivalent quantity of each, hydrogen being assumed as the unit of comparison; the second column contains the bulk or measure of the preceding weight; and the third column their specific gravities, hydrogen being in each case the standard of comparison. The succeeding figures give the composition of the compound gases by weight and measure:—

NAMES OF THE GASES.		Weight.	Bulk of preceding weight.	Sp. gra.	Composition compounds.	Weight.	Bulk.
Inflammables,	Hydrogen, ..	1	1	1			
	Hydruret of Carbon,....	7	½	14	Hydrogen, Carbon, ..	1 6	1 1
	Bihydruret of Carbon,....	8	1	8	Hydrogen, Carbon, .	2 6	2 1
	Bicarburetted Hydrogen,	13	?	?	Hydrogen, Carbon, ..	1 12	1 2
	Carbonic Oxide,	14	1	14	Oxygen, .. Carbon, ..	8 6	½ 1
	Carbon Vapour,	6	1	6			
Supporters of combustion,	Oxygen,.......	8	½	16			
	Atmospheric air,	36	2½	14⅖	Oxygen, .. Nitrogen,	8 28	½ 2
Products of combustion,	Nitrogen, ...	14	1	14			
	Carbonic acid,	22	1	22	Oxygen, .. Carbon, ..	16 6	1 1
	Sulphurous acid, ...	32	1	32	Oxygen, .. Sulphur, ..	16 16	1 1
	Aqueous vapour	9	1	9	Oxygen, .. Hydrogen,	8 1	½ 1

The process of combustion, under ordinary circumstances, cannot take place without the presence of oxygen; and the calorific value of different fuels is found to bear a close relation to the quantity of oxygen required for their complete combustion. Six parts, or one equivalent of pure carbon, require sixteen parts, or two equivalents of oxygen for combustion; and one part of hydrogen requires eight of oxygen. We may assume, as a near approximation, that two equivalents of oxygen (either by weight or measure) are required for the combustion of one equivalent of the inflammable gases derived from coal. Now, by referring to the preceding table, we find that 36 parts of atmospheric air, by weight, contain eight parts of oxygen and twenty-eight of nitrogen; and that two and a half parts by volume, contain half a measure of oxygen and two measures of nitrogen. Dalton and Prout give the weight of 100 cubic inches of atmospheric air (at 60° Fahr., with the barometer at thirty inches), at thirty-one grains Troy. We may assume, therefore, that one cubic foot of air, as usually supplied to a furnace, weighs 535 grains, and that 11·46 lbs., or 150 cubic feet, of atmospheric air, are required for the combustion of one pound of pit coal; or for each ton of coal consumed, that 11·46 tons, or 336,000 cubic feet of air are required. For the combustion of one pound of coke we may allow thirteen pounds of atmospheric air. It must be understood that these are the theoretical quantities absolutely necessary for the combustion of the fuel; whereas, in practice, the air which comes in contact with the burning mass, becomes so vitiated by nitrogen and carbonic acid, as to be incapable of supporting combustion without frequent renewal. From this cause, and the imperfect mingling of the air with the inflammable gases, a quantity equal to twice the theoretical amount should be admitted to the furnace to ensure active combustion.

It has been calculated that 100,000 cubic feet of air are required to supply the oxygen for 10,000 cubic feet of inflammable gases, contained in one ton of coals, and that, consequently, about 236,000 cubic feet are expended in supplying oxygen to the solid carbon. We have said that ten volumes of air mix with one volume of the inflammable gases, the products being eleven volumes; namely, eight volumes of nitrogen, two volumes aqueous vapour, and one volume carbonic acid. During combustion the nitrogen becomes free by the subtraction of the oxygen from the air, two volumes of hydrogen combine with one volume of oxygen to form two of aqueous vapour, and one volume of carbon combines with one volume of oxygen to form one carbonic acid. Also, during combustion, 8·12 parts, by weight, of the inflammable gases become 24·12 parts, by weight, of gaseous products; two parts of hydrogen combining with eight of oxygen to form ten of water, and 6·12 parts of carbon combining with eight of oxygen to form 14·12 carbonic acid.

In the preceding analyses we have supposed all the inflammable gases to have the constitution of the predominating one, namely, the bihydruret of carbon, more commonly called carburetted hydrogen; which will afford a sufficiently near approximation. If we assume the weight of this gas at 17 grains Troy per 100 cubic inches, then one cubic foot will weigh 292·8 grains, and 10,000 cubic feet will weigh 418 lbs. avoirdupois, which is the weight of this gas obtained by distillation from one ton of coal, or $\frac{1}{5\cdot3}$ part of the whole weight. Of the 418 lbs., 313½ lbs. are carbon, and 104½ lbs. hydrogen, or $\frac{1}{21\cdot5}$ part of the ton. As the mixed gases, however, which are given off in the furnace are somewhat heavier than we have stated above, (owing chiefly to the presence of the heavy hydruret of carbon, or olefiant gas), it will be nearer the truth if we reckon 450 lbs. as the weight of the volatile products of distillation from one ton of coal, leaving 1790 lbs. of solid carbon and earthy matter; or, subtracting 5 per cent. for ashes, we have about 1700 lbs. of carbon, consuming 230,000 cubic feet of air, at the rate of 13½ lbs. of air to 1 lb. of carbon.

Combustion considered Practically.—The carbonic acid gas resulting from the combustion of the solid matter, has a most injurious effect in checking the progress of combustion within the furnace, as it has been found by experiment that flame is barely supported in a mixture of one part of this gas with four of atmospheric air. Hence the necessity for admitting about twice the amount of air chemically required to support combustion; for, if we analyze the contents of the chimney of even a well constructed boiler, we shall find from one fourth to one half of the free oxygen present which entered the fire. The whole of the air thus supplied in excess must be heated to a very high temperature before the combustion of the inflammable gases can ensue in the furnace; and it may now be considered an established fact, that in ordinary cases of combustion in furnaces the evaporative or calorific value of a bituminous coal is nearly expressed by the calorific value of its coke; the heat of combustion of its volatile products proving in practice little more than that necessary to volatilize them, and to heat the surplus air required for their subsequent combustion.

When a stinted supply of oxygen is furnished to the incandescent fuel on the grate-bars, an imperfectly oxygenated carbonic acid, named carbonic oxide, is partly produced, accompanied by a corresponding loss in the heat generated during combustion. As this gas is inflammable, and that at a comparatively low temperature, it may be burnt in the furnace or flues of the boiler, by the admission of a further supply of atmospheric air, otherwise it escapes unconsumed up the chimney; and, in the case of marine boilers, is frequently seen burning at the top of the funnel, having caught fire on coming into contact with the oxygen of the external air. It is beneficial to avoid the formation of carbonic oxide, by the use of thin fires and plenty of air-space between the bars, so that the burning fuel may have every facility for combining with the oxygen requisite for forming carbonic acid, as it is apparent that no advantage is gained, but rather the reverse, by suffering carbonic oxide to be first formed at the fire-bars, and then burning it in the flues by admitting a rush of cold air. Hence it becomes desirable in practice to employ a large grate-bar surface for burning the fuel, which is spread out thinly over the bars. These should not exceed ¾ inch thick, and have been made ½ inch thick with advantage, the necessary strength being given them by increasing the depth. The space between the bars should be as great as possible, without incurring undue waste by the falling of cinders and small coals between them. For burning coke and wood they are made wider apart, and this is one of the causes of the very economical combustion of coke when a sufficiently strong draught can be obtained for it.

Smoke-burning Apparatus.—It is evident that a great waste of fuel must ensue from the use of smoky and badly constructed, or ill-tended furnaces, independently of the discomfort and injury to health produced by the smoke in large cities. Hence, the combustion or prevention of smoke from steam boiler or other furnaces has long engaged the attention of engineers, and has proved, besides, a most fertile field for pseudo-inventors and quacks. The quantity of solid carbon which passes off from such furnaces in soot, forms but a small portion of the actual loss; the more important part consisting of the carburetted hydrogen, and other inflammable gases, which escape in large quantities, and are not only lost in themselves, but take with them a large portion of heat which they have absorbed in acquiring the gaseous form.

The different plans which have been resorted to for the combustion of smoke and the unconsumed gases, may be divided into three classes, namely, those in which water or steam is introduced into the furnace to consume the smoke; those in which the smoke and gases are made to pass over or through the mass of incandescent fuel, from which the volatile matter has been extracted; and those in which the admission of hot or cold air into the furnace or flues, has been the means adopted. By the first of these methods, namely, the use of steam thrown amongst the volatile products of the furnace, the grosser particles of the smoke may certainly be consumed, but as certainly without producing any economical result; this plan has accordingly been abandoned. The second process, by which the smoke and mixed gases are passed through the candescent fuel, without any further supply of oxygen being vouchsafed them, is evidently very unscientific, and cannot accomplish the intended object of consuming the inflammable portion of the mixed gases, although here, as in the last case, the quantity of black smoke issuing from the chimney may be much diminished. Numerous modifications of this plan are in use, and they certainly succeed in diminishing the amount of black smoke issuing from a chimney, in cases where the malconstruction of the furnace, or ignorance and carelessness in firing, had seemed to render some such application desirable.

The third plan for burning smoke and the unconsumed gases, and the only one which can be attended with an economical result, is by the introduction of pure air into the furnace or flue. Mr. Hall, of Bassford, employs a number of cast iron pipes placed in the flues of a land boiler, into which the atmospheric air is admitted, and becomes heated by the hot gases of the flue. From these pipes, small branches pass through the flue to the front of the fire-place, and the hot air is delivered into the upper part of the furnace, where it mingles with the gases, and effects their partial combustion. For a marine boiler, a casing of iron encloses the lower part of the chimney and part of the boiler, the casing being open at the top for the admission of air, which is conducted to the front of the furnace as before. An obvious objection presents itself to heating the air in this manner, from the consequent cooling of the flues or chimney, and the diminished draft thus caused, but we think favourably of the plan of introducing the air into the upper part of the furnace.

In Mr. Wye Williams' patent, cold air is admitted to the gases at and beyond the bridge, or termination of the furnace, through a series of tubes or plates which he calls "diffusion" pipes or plates, the whole arrangement being christened an "argand furnace." This plan, with care and skilful management, may be made to work very effectively, both as regards economy of fuel and the prevention of smoke; but the question with many engineers is, whether the same amount of care and skilful management bestowed upon a well-constructed furnace of the common description, may not be productive of results equally beneficial. That the "argand furnace" requires much attention in the proper regulation of the air apertures will be evident from the following considerations. If air be admitted into the flues in such a quantity as to cause the combustion of the whole of the gaseous matter evolved, immediately after a fresh charge of fuel an immense body of flame will be produced, and the portions of the boiler exposed to it will be intensely heated, and burnt, as by the blast of a blow-pipe. But when, on the contrary, the fire on the bars has burnt clear, and little else than carbonic acid and nitrogen are passing over the bridge, the flow of cold air into the flues will greatly cool down the boiler bottom, the draft will be injured, and the steam will get low. The iron plates being frequently heated and cooled alternately are rapidly destroyed, the laminæ of the metal detaching itself in blisters, which, of course, weakens the boiler, and has been stated to have caused several explosions. The last plan we shall mention is that of Joseph Williams & Co., in which heated air is employed. In this case, the bridge at the back of the furnace is split, or, in other words, there are two bridges, one before the other; between them is placed a long cast iron box reaching from end to end of the bridge, and having a lid turning on bolts fixed into the centre of the ends, and a bent lever reaching to the front of the furnace, by which the fireman can open or close the lid at pleasure to any desired extent. Hot air from the ash-pit is admitted to this box by pipes running along the side of the ash-pit, beneath the fire bars, and thus becomes intensely heated before it mixes with the smoke and gases of the furnace.

Notwithstanding all that has been done and written with reference to smoke-burning apparatus, it is the opinion of those most conversant with the subject, that by a proper construction of the furnace, and a skilful management of the fire, the smoke and gases may be as effectually, and certainly more safely, consumed by admitting the proper quantity of air for this purpose at once through the grate-bars.

The requisites for this most important object are, first, a considerable area of fire-bar surface, so that the fires may be kept thin and open; secondly, a large "dead plate" in front of the furnace, on which the coals may be coked, previously to being thrown upon the bars, by which means the smoke and inflammable gases, being volatilized at the mouth of the furnace, are burnt by passing over the clear fire before they reach the bridge; thirdly, a good draught through the fire-bars, obtained by a chimney of sufficient height, and a proper set of dampers both under the fire doors in front of the bars, and in the chimney-flue; and fourthly —which is, perhaps, the most important requisite of all, and the one most neglected—an experienced and intelligent fireman.

Thus, in the case of the boiler experimented upon for carrying out the parliamentary inquiry on coals, already referred to, we are told that "It was found unnecessary to correct for any inflammable gases flying up the chimney, because repeated analyses of the chimney gases proved them not to contain any combustible constituent; the only products ever found being carbonic acid, sulphurous acid, oxygen, and nitrogen."

Experimental Boiler—Parliamentary Report.—It may be interesting to give the details of this boiler, which is constructed on the Cornish plan. It is cylindrical in form, 12 feet long and 4 feet diameter, with flat ends, and an internal flue 2 feet 6 inches diameter, in which the grate is placed at one end. The hot air and gases, after leaving the furnace, pass through the central flue to the other end of the boiler, where they divide and return to the front of the boiler, along the outside, by a flue on each side; the two side branches, which are 2 feet 6 inches deep, then turn down under the front of the boiler and unite in the bottom flue, which is 2 feet 6 inches wide, traverse the whole length of the boiler again, and finally enter the base of the chimney, after exposing, during a course of about 36 feet, an area of 197·6 square feet of boiler surface to the heating action. A damper is placed in the flue just before entering the chimney. The chimney is 13½ inches × 13½ inches inside, and has consequently an internal area of 182¼ square inches, and the whole height is 35 feet 6 inches. The fire grate is 2 feet 6 inches wide, and 2 feet long, thus presenting an area of 5 square feet of grate-bar surface; the bars are ¾ inch thick, and have ½ inch spaces between them. The dead-plate for the coking of the coal in front of the grate is 10 inches wide. Dampers are fitted in front of the ash pit. During the experiments, the boiler was worked under a pressure varying from 1 to 3 lbs. per square inch. In the management of the fire, care was taken to supply the coals in pieces not exceeding 1 lb. weight, and in quantities of not more than one or two shovelsfull at a time, spread evenly on the fire, except in the cases of the anthracite and bituminous coals. With anthracite it was found that the sudden application of heat caused the pieces to split and fall through the bars; hence, a gradual heating on the dead-plate was beneficial. With the bituminous coals a preparatory process of partial coking on the dead-plate prevented their coking in the fire, which would have impeded the passage of air through the grate, besides giving better opportunity for burning the smoke and gases, by passing them over a large surface of ignited fuel.

Velocity of the Draught.—As the perfect combustion of the fuel has been shown to be wholly dependent upon the amount of oxygen supplied, it is evident that the velocity with which the air passes through the fire will materially affect the result. Dr. Ure's experiments to determine the velocity with which the products of combustion pass off from the furnace, led him to adopt 36 feet per second as the initial velocity, or a mean velocity of 28 feet per second through the flues of land boilers. In marine flue boilers, where the draught is checked by the numerous sudden turns of the flues, besides the friction against the side, it is usual to allow for a velocity not exceeding 12 to 15 feet per second in proportioning the areas of the flues. We have already stated that 150 cubic feet of air are required, chemically, for the combustion of 1 lb. of coal, 45 cubic feet entering into combination with the gases, and 105 with the solid carbon. The air, after combining with the gases, is increased in volume by $\frac{1}{10}$, and becomes 49·5 cubic feet; the portion combining with the solid carbon retaining the same bulk as before,—hence, we have 154·5 cubic feet as the volume of the products in their cool state at about 60°. The mean temperature of a furnace has not been very clearly ascertained; but it will be sufficient for our purpose to assume it at 1000° Fahrenheit, by which temperature the gaseous products will have increased to about three times their original bulk, or 463·5 cubic feet. We must now allow for the surplus air required to carry on the combustion, which we have stated at nearly the same as that chemically requisite, so that, probably, 800 cubic feet of gaseous matter is not too large an estimate for the volume passing over the "bridge" for each pound of coal consumed on the grate-bars.

It is found advantageous in practice, however, to allow twenty-six square inches of area over the bridge, at the back of the furnace, for each square foot of grate-bar surface, which is a proportion considerably greater than even what we have supposed above. The most approved proportions of boilers for different purposes, with the details of their construction, will be given in the practical part of the essay.

Economic Value of Fuels.—The following table gives the mean average evaporative value of fuels, reduced from the parliamentary report on coals, and will explain itself.

TABLE SHOWING THE ECONOMIC VALUES OF COAL.

Properties.	Welsh Coals.			Scotch Coals.			English Coals.	Patent Fuels.
	Mean of nineteen samples.	Highest.	Lowest.	Mean of five samples.	Highest.	Lowest.	Mean of two samples	Mean of three samples.
Pounds of water evaporated from 212°, by 1 lb. coal,......	8·9	10·21	6·36	7·64	8·46	7·08	7·9	9·20
Economic weight of one cubic foot of the coal,............	56·15	66·16	50·5	53 0	55·0	49·8	53·5	66·5 lbs.
Theoretic weight of one cubic foot of the coal, from density,	82·0	86·7	78·3	79·2	80·48	78·6	79·0	70·66 lbs.
Space, in cubic feet, occupied by one ton, economic weight,..	40·0	44·3	33·8	42·0	44·98	40·13	41·9	33·7
Weight of water evaporated by one cubic foot of coal (economic) from 212°,....	511·2	608·7	401·3	400	460·8	352·6	423·5	617 lbs.
Rate of evaporation, or the number of pounds of water evaporated per hour,..........	411·3	529·9	247·2	401·2	465	355	442·5	475 6

3·1 lbs. of water were evaporated by 1 lb. of wood (rather damp), but this is a low average.

In well-constructed boilers, about ⅔ of the theoretical heating power of the coal become available in practice, and no more; 13½ lbs of water being about the theoretic average quantity convertible into steam, by 1 lb. of Welsh coal. Of this quantity, about 2¾ lbs. are due to the hydrogen, and 10¾ lbs. to the carbon of the coal. According to Despretz, one part, by weight, of pure carbon, combining with 2·66 parts of oxygen during combustion, will raise 78·15 parts of water from the freezing to the boiling point.

The substances produced by the destructive distillation of 100 parts of coal are as follows, taking the Craigola Welsh coal as an average sample of all:—

Tar,..............................	1·2	Sulphuretted hydrogen,.....	Traces.
Water,............................	3·1	Olefiant gas and carbon,...	0·23
Ammonia,........................	0·17	Other inflammable gases,..	7·01
Carbonic acid,..................	2·79	Coke,............................	85·5

In highly bituminous coal, in which a large quantity of the constituents are rendered gaseous during the process of destructive distillation, (which occurs in furnaces before combustion), so much heat is expended in this act that the heat developed by the after combustion of the gases is frequently not greater than that abstracted during their formation. This is the cause of the apparently anomalous fact of the nearly equal heating power of bituminous coal and of its coke. The performance of the patent fuels experimented upon is, as it will be seen from the table of economic values, very high. The most of the varieties hitherto attempted have been formed by mixing bituminous or tarry matter with bituminous coal, the effect of which is that the composition is apt to form a viscid, unmanageable mass upon the bars, preventing the free passage or admixture of air to the products of combustion, and hence accompanied generally with a dense black smoke.

From some recent experiments on the evaporative power of cylindrical boilers, by M. Cavé of Paris, we obtain a mean of 8.2 lbs. of water evaporated by 1 lb. of coal.

Mr. Wicksteed's experiments on the comparative evaporating powers of different coals, give the following results, in pounds of water evaporated by one pound of coal:—

Best Welsh Coal,.........	9·5 lbs.	Average small Newcastle,	8·07 lbs.
Anthracite,.................	9·01 —	Average Welsh, ...	8·04 —
Best small Newcastle, ...	8·52 —	Average large Newcastle,	7·66 —

In estimating the comparative evaporating powers of coals, much must, of course, depend upon the construction of the boiler, and the manner in which the fuel is burnt on the grate. Thus, in Cornwall, where every attainable advantage is given to the fuel by large heating surface, slow combustion, and prevention of loss of heat by radiation, the average evaporating value of 1 lb. of coals may be taken at 10 lbs. water; and in marine tubular boilers, at 8½ lbs. to 9 lbs. An evaporative power of about 14 lbs. of water is the result theoretically possible for a good sample of coal.

Spontaneous combustion in coal is the effect of a slow process of oxidation or decay, accompanied, as usual, with heat; the gas evolved during the process being principally carbonic acid. This dangerous action proceeds more rapidly at an elevated temperature, and where a certain degree of moisture is present; but more especially in cases where sulphur or iron-pyrites (sulphuret of iron) form a constituent part of the coal. Perfect dryness and freedom from sulphur are the best preservatives with which we are at present acquainted against spontaneous combustion in coal.

SECTION III.

HISTORICAL SKETCH OF THE INVENTION AND PROGRESSIVE DEVELOPMENT OF THE STEAM ENGINE.

It is not surprising that the force of steam, when confined in close vessels, should have been known in the most remote ages of antiquity; and we accordingly learn, that this power was early employed by the priesthood of Egypt and Greece for communicating motion and sound to the statues and images of their temples, with the view of deluding their worshippers into the belief of supernatural agency. We also read in Aristotle, Seneca, and others of the ancient philosophers, that the phenomena of earthquakes were attributed by the learned of their days, with much perspicuity, to the expansive force of the vapour of water in contact with the subterranean heat. But although the power of steam was thus recognised as the probable cause of the most stupendous force exerted in nature, we have no evidence that the ancients ever conceived the idea, or even the possibility, of measuring and controlling this power as an assistant to human labour. Until a comparatively recent date, the only applications of steam with which we have been made acquainted had for their object either the encouragement of superstition, or the amusement of philosophers, by the construction of automatic toys. Many of these show considerable ingenuity, the steam having been employed either directly by its elastic force, as in raising and propelling liquids; or by the force of impact, as in giving motion to the vanes of a little windmill; or by the reaction of a jet issuing from an orifice, as in Hero's famous toy of the Æolipile.

Hero's Æolipile, B.C. 120.—This ingenious mechanician, who flourished at Alexandria about 120 years before Christ, thus describes his apparatus, in a treatise still extant, entitled "Spiritalia seu Pneumatica:"—"Let a boiler be set on the fire and nearly filled with water, and let its mouth be closed by a cover, which is pierced by a bent tube, whose extremity fits exactly into a hollow sphere. But at the opposite end of the diameter of the sphere let there be an iron axis supported from the top of the cover; and let the sphere have two bent pipes from the ends of a diameter of the sphere perforated therewith, and bent round in opposite directions; and let the bends make right angles, and be in the plane perpendicular to the axis of rotation. Then it will follow that, the boiler being heated, the vapour, rushing through the tubes and the sphere, will rush out through the reversed pipes, and whirl the sphere round on its axis." It is apparent that this instrument derives it rotatory motion from the reaction or recoil exerted at the extremities of its two arms, by the rapid issuing of the steam; and it is identical in principle with Mr. Ruthven's modern engine, which, upon the whole, is an effective, and certainly a very simple, arrangement, though by no means an economical one as regards the consumption of steam.

Blasco de Garay, A.D. 1543.—The first historical notice we possess of the application of an elastic fluid to useful purposes dates from the year 1543, when we are informed that Blasco de Garay, a sea-captain in the service of Charles V. of Spain, succeeded in propelling a ship of 200 tons burthen in the harbour of Barcelona, at the rate of a league (or three miles) an hour. No information is afforded of the nature of his apparatus, except that it comprehended a boiler, which, it is stated, was liable to burst—that the power was transmitted through paddle-wheels—and that the vessel could be put about with much facility, by means of the apparatus. We can only speculate on the nature of this mysterious engine, but it seems probable that it was propelled by high-pressure steam, on the principle of the Æolipile. Notwithstanding that the scheme was partially approved by the emperor and his ministry, and its author promoted, we do not read of any second experiment being made, or of any farther notice being taken of the invention.

Solomon de Caus, A.D. 1615.—In the year 1615, a work entitled "Les Raisons des Forces Mouvantes avec diverses Machines tant utiles que plaisantes," was published at Frankfort, by Solomon de Caus, a Frenchman. In this book some simple apparatus is described for elevating water in a tube, and bursting copper balls, by the agency of fire; but it is remarkable that no mention is made of the vapour of water as an elastic gas *in propriâ personâ*, but only when in combination with atmospheric air—the latter being evidently considered the chief agent in producing the results specified. It is at least apparent that the author's ideas upon the subject of steam-pressure were very obscure, not to say erroneous; and we are at a loss to discover upon what grounds M. Arago claims for De Caus any part in the invention of the steam engine. He writes thus, in treating of air, "The violence will be great when water exhales in air by means of fire, and that the said air is enclosed; as for example, take a ball of copper, one or two feet in diameter and one inch thick, which being filled with water by a small hole, afterwards strongly stopped with a peg, so that neither air nor water can escape, it is certain that if we put the said ball upon a great fire, so that it will become very hot, that it will cause a compression so violent, that the ball will burst in pieces with a noise like a petard." Elsewhere he explains the process of evaporation and condensation by saying, that "the parts of the elements of air and water mix together for a time, and then each returns to its place,"—a very crude and imperfect definition.

Giovanni Branca, A.D. 1629.—In 1629, Giovanni Branca, an Italian, published a work in which, amongst other mechanical inventions, he describes and illustrates a machine propelled by high-pressure steam. It consists of a large wheel furnished with flat vanes on the periphery, against which the steam is directed in a jet, causing the wheel to revolve by the mere impact of its particles. This scheme possesses little merit or originality; but it is somewhat redeemed by the suggestion of several useful applications of the power,—such as driving pestles and stones for grinding gunpowder, machines for raising water in buckets, &c. It is most probable that Branca's scheme was never reduced to practice; for had it been so, it would have been found a most inefficient mode of generating power, and much inferior to Hero's Æolipile.

Marquis of Worcester, A.D. 1663.—We now arrive at the period of the Marquis of Worcester's invention, which is the first to which the name of a steam engine can properly be applied. This is described by him in a work published in 1663, entitled "A Century of the Names and Scantlings of Inventions," where, under the head of the 68th invention, we read as follows:—

"A Fire-Water Work. An admirable and most forcible way to drive up water by fire, not by drawing or sucking it upwards, for that must be, as the philosopher calleth it, '*infra sphæram activitatis*,' which is but at such a distance. But this way hath no bounder if the vessels be strong enough; for I have taken a piece of a whole cannon, whereof the end was burst, and filled it three-quarters full of water, stopping and screwing up the broken end, as also the touch-hole, and making a constant fire under it; within twenty-four hours it burst and made a great crack. So that having a way to make my vessels so that they are strengthened by the force within them, and the one to fill after the other, I have seen the water run like a constant fountain stream forty feet high. One vessel of water rarified by fire driveth up forty of cold water; and a man that tends the work has but to turn two cocks, that, one vessel of water being consumed, another begins to force and re-fill with cold water, and so successively, the fire being tended and kept constant, which the self-same person may likewise abundantly perform in the interim between the necessity of turning the said cocks." The Marquis says, in another place, that "his waterwork is by many years' experience and labour so advantageously contrived, that a child's force bringeth up, an hundred feet high, an incredible quantity of water, even two feet diameter, so naturally, that the work will not be heard into the next room, and with so great ease and geometrical symmetry, though it work day and night from one end of the year to the other, it will not require forty shillings' reparation to the whole engine, nor hinder one day's work; and not only with little charge to drain all sorts of mines and furnish cities with water, though never so high seated, as well as to keep them sweet, running through several streets, and so performing the work of scavengers, as well as furnishing the inhabitants with sufficient water for their private occasions."

We cannot infer from the foregoing descriptions that Lord Worcester employed any other means than the simple elastic force of the steam in raising water, although he was evidently not ignorant of the principle of atmospheric pressure; so that his merit lies not in the discovery of any new principle, but in the ingenious adaptation of known means to the construction of an original apparatus, which in magnitude and efficiency leaves all its predecessors far behind. Its chief fault lies in the rapid condensation of the steam, which must have ensued on its first coming into contact with the surface of the cold water and the sides of the receiver; this hurtful and useless condensation proceeding until a film of hot water had been formed on the top. It is quite probable that such an engine would perform all the work ascribed to it, but the consumption of fuel inferred from the foregoing extracts is enormous, being about 200 times greater than in steam engines of the present day.

Sir Samuel Morland, A.D. 1683.—Sir Samuel Morland was master of the works to Charles II. in 1680, and whilst on a professional visit to France, in 1683, he published a treatise in the French language, entitled "L'Elevation des Eaux par toute sorte de Machines, reduite à la Mesure, au Poids, et à la Balance." In this the following passage occurs:—"Water being converted into vapour by the force of fire, these vapours shortly require a greater space (about 2000 times) than the water before occupied, and sooner than be constantly confined would split a piece of cannon. But being duly regulated according to the rules of statics, and by science reduced to measure, weight, and balance, then they bear their load peaceably, (like good horses); and thus become of great use to mankind, particularly for raising water, according to the following table, which shows the number of pounds that may be raised 1800 times per hour to a height of six inches by cylinders half filled with water, as well as the different diameters and depths of the said cylinders." In this general description we discover no advance upon Lord Worcester's invention;

but it is remarkable from the near approximation which Morland gives of the actual expansion of water in its conversion into steam.

Dr. Papin, A.D. 1690.—About the year 1690, Dr. Papin, a native of Blois, in France, first suggested the employment of a floating piston, working within a cylinder, as a means of raising water through the agency of steam. His idea, (for it is believed never to have been put in practice), was to have a cylinder, with water in the bottom, which the fire was to be alternately applied to, and removed from; the *ascent* of the piston being effected by the elasticity of the steam and its *descent* by the pressure of the atmosphere against the partial vacuum produced, in process of time, by the gradual condensation of the steam underneath. Although this scheme can be considered only in the light of a theoretical suggestion, still Papin deserves much credit both for his idea of the cylinder and piston, and the condensation of steam to effect a vacuum; and it will be seen that before long both of these suggestions were acted upon with much advantage by more practical men.

Capt. Savery, A.D., 1698.—Captain Savery's patent, granted in the year 1698, (being the first on record for an invention of this description,) forms the next link in the progressive development of the steam engine. His apparatus, which was mainly directed to the raising of water from mines, may be thus described: Two boilers were employed, the one for generating steam, and the other, which was much smaller, for feeding the large boiler with hot water. This was done by means of a connecting pipe, through which the feed water was forced by an excess of pressure in the small boiler. In starting the engine, the steam is allowed to enter a spherical copper vessel or receiver, until it has driven out all the air, and has ceased to be condensed against the cold sides, technically called "blowing through." The communication with the boiler is then closed, and a jet of cold water is thrown upon the outside of the receiver. The contained steam is thus partially condensed and a vacuum formed, the water, meanwhile, rising into the receiver by the external pressure of the atmosphere. A communication is again opened with the boiler, and the steam pressing on the surface of the water in the receiver forces it up a main-pipe to a height proportional to its elasticity, its return in the pipe being prevented by a valve opening upwards. By employing a double set of receivers to draw and force the water alternately, a constant ascending stream may thus be maintained. The boilers were fitted with water gauge cocks for ascertaining the level of the water, and with the steel-yard safety valve which had been previously invented by Dr. Papin for his digester. Savery's engine, although displaying much ingenuity both in the principles and in the working details of its construction, could never have been very effective in consequence of two disadvantages, which, at the present day, are sufficiently obvious. These are the wasteful condensation of the steam when it enters the cold receiver and presses on the surface of the cold water therein; and the limited height to which the water could be forced by the elasticity of the steam, which was thus brought into direct competition with the height of the column, an additional pressure of 15 lbs. in the boiler being requisite to balance every thirty-three feet of water.

Newcomen, A.D., 1705.—These disadvantages soon led to the abandonment of Savery's engine, for one invented by Thomas Newcomen and John Cawley, mechanics of Dartmouth. Their apparatus, although not remarkable for its mechanical ingenuity, possessed the merit of being the first to embody Papin's suggestion of the cylinder and piston, which soon brought about a revolution in the construction of the steam engine. As Captain Savery's method of producing a vacuum by the condensation of steam was an essential feature in the new engine, he was also associated in the patent, which was taken out in 1705. By the use of the cylinder and piston, two grand improvements upon Savery's engine were effected,—the steam was not condensed by contact with the surface of the cold water, and the height of the column of water to be raised became independent of the pressure of steam in the boiler, provided the diameter of the cylinder was increased proportionably to the load. Newcomen wisely resumed the use of pumps, which he connected with the piston-rod of his cylinder by means of a working beam placed over the engine. The cylinder was open at the top, and was supplied with steam of low tension, by a pipe leading from the boiler to the bottom of the cylinder, the valves being worked by hand. In starting the engine, the steam was admitted under the piston until the cylinder was heated, and the air had been expelled through the "snifting" or "blow-through" valve in the cylinder bottom. The piston may be supposed to be at the top of the cylinder, which is full of steam; the communication with the boiler was then closed, and cold water allowed to run into an external casing surrounding the cylinder, by which means the steam was condensed, and the piston descended by the external pressure of the atmosphere. The ascending stroke was next performed by re-admitting the steam under the piston, to balance the atmospheric pressure, the weight of the pump-rods at the other end of the oscillating beam sufficing to raise the piston to the top of the stroke. A little water was constantly kept over the piston to assist in rendering it steam and air tight; and a small cock was fitted to the bottom of the cylinder for drawing off the water resulting from condensation, and leakage through the piston, which was employed in feeding the boiler. Such was the primitive construction of Newcomen's atmospheric engine, but it was soon afterwards improved by the accidental discovery of condensation by injection within the cylinder, and by a method of working the valves from the moving parts of the engine itself. The manner in which these improvements were effected is thus related by Desaguliers in his Experimental Philosophy:—"In the latter part of the year 1711, Newcomen and Cawley made proposals to drain the water of a colliery at Griff, in Warwickshire, where the proprietors employed 500 horses, at an expense of L.900 a year; but their invention not meeting with the reception they expected, in March following, through the acquaintance of Mr. Potter of Bromsgrove, in Worcestershire, they bargained to draw water for Mr. Back of Wolverhampton;

where, after a great many laborious attempts, they did make the engine work; but not being either philosophers to understand the reason, or mathematicians enough to calculate the powers and proportions of the parts, they very luckily, by accident, found what they sought for.

"They were at a loss about the pumps, but being so near Birmingham, and having the assistance of so many admirable and ingenious workmen, they came, about 1712, to the method of making the pump valves, clacks, and buckets, whereas they had but an imperfect notion of them before. One thing is very remarkable: as they at first were working, they were surprised to see the engine go several strokes, and very quick together, when, after a search, they found a hole in the piston, which let the cold water in to condense the steam in the inside of the cylinder, whereas, before, they had always done it on the outside. They used before to work with a buoy to the cylinder, enclosed in a pipe, which buoy rose when the steam was strong, and opened the injection and made a stroke; thereby they were capable of giving only six, eight, or ten strokes in a minute, till a boy named Humphrey Potter, in 1713, who attended the engine, added (what he called *scoggan*) a catch that the beam always opened, and then it would go fifteen or sixteen strokes a minute. But this being perplexed with catches and strings, Mr. Henry Beighton, in an engine he had built at Newcastle-upon-Tyne, in 1718, took them all away but the beam itself, and supplied them in a much better manner."

The next improvement to the engine was the addition of the cold water pump, worked from the reciprocating beam, to raise the water required for injection above the level of the cylinder. This water, after having done its duty in the cylinder, was expelled, along with the condensed steam, through an escape-valve, into the feeding cistern, by the descent of the piston.

A.D., 1718.—Beighton's improvements on the atmospheric engine date from the year 1718, from which time this engine continued in use, with but little alteration or improvement, for half a century. Brindley, indeed, applied an improvement to the feeding apparatus, which he made self-acting, by means of a float on the surface of the water in the boiler, communicating by a wire with a valve in the feed-pipe, which was thus opened whenever the water in the boiler fell below a certain level.

SMEATON, A.D., 1767.—In the year 1767, Smeaton's attention was first directed to the steam engine from having been employed to construct an atmospheric engine for the New River Company. Being of opinion that power was lost by the too rapid reciprocation of the beam and the other moving parts of the engine, since the whole momentum acquired during the stroke was necessarily annihilated at each change of direction in the motion, he resolved to increase the length of stroke, and remove the centre of the working beam nearer the pump-rod end, employing larger pumps to compensate for the slowness of the motion, and loading the piston with 10½ lbs. per square inch in place of 7 lbs., which had been the usual practice. One of the chief advantages he contemplated from these alterations was a diminution in the amount of injection water required to condense the steam, but in this he was disappointed; finding, to his surprise and mortification, that more injection water was necessary than before, and that the cylinder was consequently so much cooled, that much useful effect was lost by the condensation of the steam on first entering it. He accordingly altered the fulcrum of the beam until the load upon the piston was reduced to 8½ lbs. per square inch. "With this load the engine went so much quicker, as not only to raise more water but consume less coals; it took less injection water, the cylinder became hot, and the injection water came out at 180° Fahr., and the engine, in every respect, not only did its work better, but went more pleasantly."

With the view of pursuing these investigations, he erected a small experimental engine, of about one horse power, at Austhorpe, from which he obtained the following results:—the consumption of coals was 55 lbs. per hour, equivalent to a duty of about 2,920,000 lbs. raised one foot high by a bushel of coals—the injection water required was 95 cubic inches per stroke, and the water evaporated from the boiler about 8·9 cubic inches per stroke. He came to the conclusion that the most beneficial load on the piston was 7·8 lbs. per square inch. Smeaton ascertained about this time the performance of most of the atmospheric engines then in use, and of fifteen at Newcastle the best was found to raise about seven and a half millions of pounds one foot high by the consumption of a bushel of coals, and the worst three and a fourth millions. The engine he afterwards erected at Long Benton, in which his improvements were fully carried out, raised nine and a half millions.

From what we have said it will appear that Smeaton introduced no material innovation in the principles of the atmospheric engine, his improvements being confined chiefly to the introduction of more accurate workmanship and juster proportions than heretofore; by which means alone he was enabled to effect a saving of probably 1½ in the consumption of fuel. He adopted the plan of clothing his cylinder and piston with a non-conducting substance, to check the radiation of heat, and he enlarged the steam-passages to the cylinder, by which the "throttling" of the steam was prevented during its passage from the boiler. His boilers were constructed on sounder principles, and were calculated with a much greater evaporative power, than any which had been employed before his time. They were of cast iron plates, bolted together by flanges, and were usually of the "haystack" form, hemispherical at top, and arched internally at the bottom; the flame being conducted, by the aid of flues in the setting, along the bottom and round the sides and ends, as in our modern boilers of this kind. Wrought iron plates were not used in the construction of boilers until a much more recent period, cast iron and copper having been the materials employed; and in several of Newcomen's old engines, the dome of the copper boiler was sometimes formed of lead.

The speed of Smeaton's engines was regulated by the contrivance called a cataract, the parent of the modern mechanism of the same name, and which had been applied to the mines in Cornwall some time previously. The apparatus is very simple, consisting in a small vessel balanced on a centre, so that when empty it stands erect, but when filled with water by the dripping of a small cock, it is over-balanced, and in its fall raises a lever, which opens the injection cock, and causes the engine to make a stroke. The speed of the engine is thus determined by the number of times that the vessel fills and empties itself per minute. In estimating the value of Mr. Smeaton's improvements, we should not forget that several of the mechanical inventions which he first applied to the air-pump, were adapted by subsequent engineers to the steam engine,—such as the close cylinder-cover and stuffing-box, &c.

James Watt, A.D. 1763.—In the year 1763, James Watt, then a philosophical instrument maker in Glasgow, first devoted the powers of his master-mind to the study and improvement of the steam engine, which thenceforth rapidly progressed under his guidance to a state of perfection little short of what we find it at the present day. His improvements merit our admiration the more that they were, for the most part, elicited by a patient and progressive course of reasoning, submitted for verification to the test of practice; and they exhibit, in a remarkable degree, the rare combination of true science with that practical and persevering turn of mind, which is requisite for adapting the abstruse theories of the philosopher to their noblest use —the service and amelioration of mankind.

Separate Condensation.— Watt's attention was first directed to the steam engine from being employed to repair a working model of a Newcomen engine, for the natural philosophy class of the Glasgow university. His experiments proved that the quantity of steam consumed by this engine was vastly more than what was effectively used in the cylinder. This loss he attributed partly to the small size of the engine, in consequence of which the cylinder presented a large comparative surface for radiation; but more especially to the useless condensation of the steam by contact with the sides of the cylinder, cooled down as it was by the injection water at each stroke. He therefore conceived the happy idea of condensing the steam in a *separate vessel*, which could always be kept cold without prejudice to the temperature of the cylinder. He surrounded the latter, at the same time, with a hollow cast iron casing, or "jacket," into which steam was suffered to flow from the boiler. The condenser, on the other hand, was kept as cold as possible, being generally immersed in a cistern of cold water.

The Air-Pump.—This arrangement suggested the adoption of the *air-pump* to withdraw from the condenser the injection water, with its contained air, the water of the condensed steam, the air from leakage, and the uncondensed watery vapour, all of which had been formerly expelled through the snifting-valve, by the descent of the piston. In Watt's earlier engines he employed condensation by surface, in order to save the work of the air-pump in lifting the injection water, but this was afterwards abandoned for the better method of condensation by injection. Dr. Roebuck, of the Carron iron-works, having heard of Watt's improvements, offered to join him in a patent, on condition of being allowed two-thirds of the profits. This was willingly agreed to, and in the year 1769, a patent right was applied for and obtained for fourteen years. The specification of this patent, taken out in Watt's name, was as follows:—

"My method of lessening the consumption of steam, and consequently of fuel, in fire engines, consists of the following principles:—

"First, That vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in common fire engines, and which I call the steam-vessel, must, during the whole time the engine is at work, be kept as hot as the steam that enters it; first, by enclosing it in a case of wood, or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and, thirdly, by suffering neither water or any other substance colder than the steam to enter or touch it during that time.

"Secondly, In engines that are to be worked wholly or partially by the condensation of steam, the steam is to be condensed in vessels distinct from the steam-vessels, or cylinders, although occasionally communicating with them; these vessels I call condensers; and, whilst the engines are working, these condensers ought at least to be kept as cold as the air in the neighbourhood of the engines, by application of water or other cold bodies.

"Thirdly, Whatever air or other elastic vapour is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the steam-vessels, or condensers, by means of pumps, wrought by the engines themselves, or otherwise.

"Fourthly, I intend, in many cases, to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the same manner as the pressure of the atmosphere is now employed in common fire engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only, by discharging the steam into the open air after it has done its office.

"Lastly, Instead of using water to render the piston, or other parts of the engines, air and steam-tight, I employ oils, wax, resinous bodies, fat of animals, quicksilver, and other metals in their fluid state.

"And the said James Watt, by a memorandum added to the said specification, declared that he did not intend that anything in the fourth article should be understood to extend to any engine where the water to be raised enters the steam-vessel itself, or any vessel having an open communication with it."

Partnership with Boulton, A.D. 1773.—Much difficulty was experienced at first from the impossibility of procuring accurate workmanship, and up to the year 1773, little progress had been made in the manufacture of the engines. In that year Dr. Roebuck's affairs became embarrassed, and he sold his interest in the patent-right to Mr. Boulton, of Soho, Birmingham, who at that time possessed the most celebrated metal works in the kingdom. An extended term for the patent was then applied for, and obtained, for twenty-five years from that date. Watt had now at his command the resources of a large establishment, by means of which he could secure the best workmanship at that time available; and the new firm consequently turned out many excellent engines within a few years. As their earlier productions were employed exclusively in pumping water, they were constructed on what is termed the single-acting principle; and in this respect only differed from Newcomen's by using the elasticity of steam, in place of the gravity of the atmosphere, for pressing down the piston in the descending stroke; the ascending stroke being performed in both cases alike by the weight of the pump-rods, or other counterbalance. By introducing the steam above as well as below the piston, the cylinder was preserved from the cooling effect of the external air; and at the same time, an additional force could be obtained, by employing steam above the atmospheric pressure. This arrangement necessarily required the use of a close cover and "stuffing-box" to the cylinder, through which the piston-rod might work steam-tight. The valves, which were conical spindle-valves, worked by tappets on the plug-beam, were situated in a steam-passage, extending the whole length of the cylinder, and admitted a constant flow of steam above the piston, and alternate steam and vacuum below. At this time, Boulton and Watt charged for their engines one-third part of the saving of fuel as compared with the old atmospheric engine; this sum to be paid annually during the term of their patent, or redeemable at a ten years' purchase. They calculated on raising 500,000 cubic feet of water one foot high by the consumption of 112 lbs. of Wedensbury coals, which is equivalent to about twenty-three and a half millions of pounds raised one foot high by a bushel of 84 lbs. This estimated "duty" was rather more than what was obtained in practice, their best engines seldom exceeding twenty millions. We may remark that the duty of the best Cornish engines of the present day exceeds one hundred millions.

Discovery of Expansion.—The next step towards the improvement of the steam engine arose from Watt's discovery of the advantages to be derived from expanding the steam in the cylinder; his researches on this subject being much assisted by Dr. Black's contemporaneous discovery of the theory of latent heat, which has been already explained. By an acquaintance with the principles of mechanics, we learn that if steam be supplied to the cylinder of an engine at a uniform density during the whole length of the stroke, the resistance being at the same time uniform, the piston will move with a continually increasing velocity to the end of the stroke, when its momentum will be suddenly and entirely destroyed before commencing the next change of direction. This necessarily occasions a serious loss of power, as well as a destructive jar to the parts of the machine. As the readiest means of checking this injurious velocity, the steam-valve was shut a little before the end of the stroke, so that the piston reached the bottom of the cylinder under a diminished pressure. By further extending this principle of cutting off the steam at a fraction of the stroke, it was found that the force actually exerted within the cylinder was much greater than what might have been expected to result from the simple pressure of the inclosed steam. This additional force, then, was due to the effort of the isolated steam, whilst expanding into the increased volume as the piston descended in the cylinder.

Principle of Expansion.—To make this subject intelligible, let us suppose a cylinder of 1 square foot area, and 20 inches long, to which steam of the atmospheric pressure has been supplied during half the stroke. We will suppose the pressure on the piston to equal one ton at the moment that the steam-valve is closed, the space under the piston being in communication with the condenser, and therefore a vacuum; but as the piston descends, the steam above it will be diffused through an increased volume, and will consequently acquire a diminished pressure. We may assume that this diminution of pressure follows the law of elastic fluids in general; that it will be decreased in the same proportion as the volume of the steam is augmented. While the piston, therefore, moves downwards from the centre of the cylinder (at 10 inches), it will be urged by a continually decreasing force, until it arrives, we will suppose, at 15 inches, when the space occupied by the steam will be increased in the proportion of two to three. The pressure on the piston will also be diminished in the inverse ratio of three to two, so that the pressure being one ton at 10 inches from the top, will be $\frac{2}{3}$ of a ton at 15 inches. In like manner, when the piston arrives at the bottom, the space occupied by the steam being double that which it occupied when the piston was at half stroke, the pressure of the steam will be diminished one-half, and consequently the pressure on the piston at the end of the stroke will be half a ton. If we calculate, in the same way, the pressure on the piston at each inch of the space through which we suppose the steam to act expansively, we shall find that after passing the first inch the volume of the steam is increased in the proportion of 10 to 11, and hence its pressure will be diminished in the proportion of 11 to 10, equalling at the eleventh inch $\frac{10}{11}$ of a ton. In like manner, at the twelfth inch it will equal $\frac{10}{12}$, at the thirteenth inch $\frac{10}{13}$ of a ton; and so on to the twentieth inch, where it will be $\frac{10}{20}$, or $\frac{1}{2}$ ton.

Now, if the pressure of the steam through each of these ten divisions were to continue uniform, and, instead of gradually diminishing, to suffer a sudden change in passing from one division to another, then the mechanical effect produced would be obtained by taking the mean of the several pressures due to each of the ten divisions. In the present case it has been supposed, that the force on the piston at the beginning of the first division was 2240 lbs. To obtain the pressure in pounds corresponding to each of the successive divisions, it will therefore be necessary to multiply 2240 by 10, and to divide it successively by 11, 12, 13, &c. The pressures, in pounds, at each inch below the half stroke, will then be as follows,—viz.,

1st,	2036·3 lbs.	6th,	1400·0 lbs.
2d,	1866·6 —	7th,	1317·6 —
3d,	1723·1 —	8th,	1244·4 —
4th,	1600·0 —	9th,	1179.0 —
5th,	1493.3 —	10th,	1120.0 —

If the mean of these be taken, by adding them together and dividing by 10, it will be found to be 1498 lbs. for the mean pressure of the steam during the expanding half of the stroke; this being nearly three-fourths of the mechanical effect exerted during the first half of the stroke by the full steam. In the preceding calculation we have assumed the pressures through each of the ten divisions to be uniform; which, strictly speaking, they are not, owing to the expansive action which takes place in each division. The more accurate computation (which involves the use of the higher mathematics) would give the average pressure for the expanding half of the stroke at 1545 lbs. It is evident that this principle will be equally applicable at whatever part of the stroke the steam may be cut off, a greater mechanical effect being obtained from a given amount of steam in proportion to the extent to which the expansive action is carried, and also in proportion to the original density of the steam. Thus, in the Cornish engines of the present day, it is found most advantageous to employ steam of 30 or 40 lbs. pressure, and to cut it off in the cylinder at one-sixth, or even one-eighth of the stroke, the remaining seven-eighths being performed by expansion. The practical limit to this principle is imposed by the increased size required for the cylinder, and the inequality in the speed of the piston during the stroke. It will be found by calculation that if the steam be cut off at one-half stroke, its effect in the cylinder will be multiplied by 1·7 nearly.

If at one-third,	2·1	nearly.
— one-fourth,	2·4	—
— one-fifth,	2·6	—
— one-sixth,	2·8	—
— one-seventh,	3·	—
— one-eighth,	3·2	—

Rotatory Motion.—The only purpose to which the steam engine had as yet been applied was the drainage of mines, for which the reciprocating rectilinear motion obtained directly from the piston-rod of the atmospheric engine was all that was required. Before its use could be extended to other purposes, however, it became necessary to devise some means for changing the rectilinear into a rotatory motion. This was first effected in 1762, by making use of the clumsy expedient of an ordinary pumping engine to raise and return the water constantly to the buckets of a water-wheel, from the axis of which a circular motion was obtained for drawing coals out of a pit. This having been soon abandoned as cumbrous and expensive, the next attempt made was to employ two endless chains, passing over pulleys, and so arranged that the joint-pins of the links of the two chains worked in the teeth of a cogged wheel, to which they imparted a circular motion; first by one chain, and then by the other, as they acted alternately on opposite sides of the wheel, the one chain impelling it during the *descent* of the piston, and the other during the *ascent*.

Fly-Wheel and Crank.—The fly-wheel now began to be applied to the steam engine as a means of facilitating and regulating the transmission of power. This it effects by virtue of the *vis inertiæ* of its heavy rim, which when once put in motion continues by its momentum the circular direction of the crank, or other mechanism, during the periods of top and bottom stroke, when the piston is at rest, besides acting beneficially to equalise temporary irregularities in the resistance offered to the piston by a varying load. The system of ratchet-work mentioned above was soon after relinquished for the common crank; which, in conjunction with the fly-wheel, has ever since held its ground as the most simple and efficient means for changing a rectilinear into a rotatory motion.

Sun and Planet-Wheels.—Although Mr. Watt claimed the merit of having first introduced this application of the crank, a patent had in the meantime been taken out for it by another party; so he was obliged to substitute for it, in his engines of that date, a more complicated arrangement called the "sun and planet wheels," which may be thus described:—A toothed wheel, called the sun-wheel, was fixed on the axle of the fly-wheel to which rotation was to be imparted. The other toothed wheel, called the planet-wheel, had an equal diameter, and was fixed on the end of the connecting-rod, so as to be incapable of revolving. During the descent of the piston, the working end of the beam will be drawn upwards, and the lower end of the connecting-rod, with its fixed spur-wheel in geer with that on the axle of the fly, will drive round the latter (we shall say from east to west), during the ascent of the connecting-rod. The planet-wheel, having now arrived at the top, will continue to travel round the sun-wheel during the descent of the connecting-rod, thus constantly driving it, with the fly-wheel attached, in the same direction. This mechanism is inferior to the crank from its being more liable to derangement, and subject to much wear and tear, but it possesses this advantage—that the planet-wheel communicates a double velocity to the fly-wheel for each revolution.

Invention of the Double-acting Engine.—It now became a matter of the utmost importance that the motion of the atmospheric engine should be rendered as equable and uniform as possible. It will be remembered that the engine was still what is called single-acting; that is to say, the whole power was obtained during the descending stroke, the ascending stroke being performed by means of a counter-weight. This necessarily made the rotatory motion very unequal, even with the assistance of a heavy fly-wheel; and it occurred to Watt that it might be advantageous to employ two engines acting upon two cranks, fixed on the same axis, at an angle of 120° to each other, with a weight placed on the circumference of the fly-wheel, at the same angle to each of the cranks. This would undoubtedly have equalised the motion on the same principle as our double marine engines of the present day, but Watt soon perceived that the same object might be better and more simply effected by using one cylinder only, in which the steam and the vacuum should act alternately on each side of the piston, the ascending and descending stroke being alike performed by the pressure of steam against a vacuum. Hence the invention of the double-acting engine, for which a patent was obtained in 1782. Thus, by a simple change in the construction and operation of the valves, a double effect was obtained from the same engine; though not, of course, without expending nearly a double amount of steam. The advantages of the double-acting engine are obvious, viz.—a more equable motion for the piston; a great comparative saving of expense, of space and of weight, in the apparatus; greater economy of the steam-power, owing to a diminished loss of effect from cooling and leakage within the pipes and cylinder; and a saving in the resistance arising from the friction of the moving parts. The work of the air-pump was but little diminished, as the injection water was now allowed to enter the condenser, in a constant, in place of an intermitting, jet.

Parallel Motion.—As the piston-rod of the new engine was now subjected to a *thrust* as well as a *pull*, it became necessary to devise some means for guiding the head of the piston-rod in a vertical position. Watt effected this object by means of that most ingenious and beautiful combination of levers called a parallel motion, which has ever since been employed to form the connecting link between the head of the piston-rod and the extremity of the sway beam. As the construction and principle of parallel motions are fully illustrated in another part of this work we shall not enlarge upon them here, but refer the reader to Plates XIII., XIV., with its description, and to page 31 of the present article.

Throttle Valve.—The next addition made to the steam engine was the "throttle valve," which was situated in the steam-pipe where it enters the valve box, and which was employed for regulating the admission of steam to the cylinder in proportion as the load on the piston was increased or diminished.

Governor.—The throttle valve was at first opened or closed by hand, but at this stage Watt invented the beautiful contrivance of the governor, by means of which the working of the throttle valve became an automatic operation dependent upon the speed of the engine. The governor will be found amply detailed, in various forms, in Plates XC. and CVIII., with their accompanying descriptions. The working beam and the connecting rod began now to be formed of cast-iron in place of timber, and the engine altogether assumed more of the appearance of the modern long-beam condensing engine.

Operation of the Double-acting Engine.—Having considered each of Watt's improvements separately, let us now briefly review the operation of his double-acting engine as a whole. The steam from the boiler, after passing through the throttle valve, whose aperture was regulated by a system of levers connected with the governor, entered the upper valve box, which contained two conical spindle valves, the one for admitting steam above the piston during the descending stroke, and the other for leading the steam away through the eduction pipe, into the condenser, during the ascending stroke. The upper valve box communicated by a steam passage with the lower valve box, in which were also situated two valves corresponding to those in the upper box. The valves were opened and shut at the proper time by catches on the plug-beam, the upper steam and lower eduction valves being open at the same time, and *vice versâ*. The steam valve was always closed a little before the end of the stroke by giving the requisite *lead* to the valve, and if it was desired to work more expansively, the steam was cut off at any fraction of the stroke by means of a distinct valve placed in the steam-pipe. The principle of expansion, however, does not seem to have been duly appreciated at that time, owing probably to the low tension of the steam employed, which did not admit of this principle being carried out to its greatest advantage. The steam, on leaving the cylinder, rushed through the eduction pipe into the condenser, where it was rapidly condensed on coming into contact with the jet of injection water, which was kept constantly playing. Meanwhile, the air-pump, which was worked by a long rod hanging from the beam of the engine, extracted the heated injection water and the uncondensed gases from the condenser, and delivered them into the *hot well*, from which a portion of hot water was extracted at each stroke by a small *feed pump*, (also worked by the engine), for supplying the boiler. The amount of injection water is regulated by hand, by means of a cock and index plate, and should be in proportion to the quantity and temperature of the steam which passes into the cylinder. The condenser and air-pump were placed in a cistern constantly supplied with cold water by the *cold water pump*, which was worked by the engine, and lifted just sufficient water for supplying injection to the condenser. The velocity of the engine was determined by the control which the governor possessed over the throttle valve, and the uniformity of the motion was ensured by a heavy fly-wheel.

Such was the steam engine as improved—we might almost say created—by the inventive genius of Watt, which has left but few gleanings for his successors in the path of advancement. There are still a few names, though of minor importance, which deserve to be noticed in an inquiry of this kind, but to these we can allude only in a very cursory manner.

Hornblower, A.D., 1781.—In 1781, Mr. Jonathan Hornblower took out a patent for a new mode of applying the expanding force of steam. With this view he made use of two cylinders, "Employing the steam, after it had acted in the first vessel, to operate a second time in the other, by permitting it to expand itself, which he did by connecting the vessels together, and forming proper passages and apertures, whereby the steam might, at proper times, go into and out of the said vessels." It can be easily demonstrated that this plan possesses, in ordinary cases, no advantage over the common method of expanding in the cylinder. It is sometimes used, however, in Cornwall, at the present day, when it is wished to carry the principle of expansion to its utmost limits.

Cartwright, A.D., 1797.—About the year 1797, Dr. Edmund Cartwright constructed a single-acting engine, in which the condenser was formed by a narrow space left vacant between two metal cylinders, one within the other, and surrounded on all sides by cold water; thus forming an ingenious and tolerably effective mode of condensation by surface. He had also the higher merit of introducing, in this engine, a metallic packing for the piston, in place of the simple hemp packing hitherto used. His packing was formed of metal rings, of the diameter of the cylinder, cut into segments, and forced by the action of springs against the internal surface of the cylinder, by which means the piston was able to adapt itself to any slight inequalities of surface.

Murray and Murdoch.—At this time, Mr. Matthew Murray, of the firm of Fenton, Murray, & Wood, made several improvements in the practical details of the engine; and about the same time Mr. William Murdoch, a partner in the firm of Boulton, Watt & Co., effected an important alteration in the steam valves, by connecting together the upper and lower valves so as to work with one rod or spindle, the tube connecting them being hollow, and serving as the eduction pipe to the upper end of the cylinder.

Woolf, A.D., 1804.—In 1804, Arthur Woolf took out a patent for a double-cylinder engine, on the principle of Cartwright's, whose patent had now expired. He employed high-pressure steam in a small cylinder, which was allowed to expand into a larger one, and was finally condensed by an arrangement similar to Watt's apparatus.

Of late years, several minor improvements have been effected in the arrangement and construction of the parts of the condensing engine, and much has been done by experience, by improved workmanship, and by the extended use of machines or "tools," to supersede manual operations. The expansive force of steam has also been applied, since Watt's time, in a more efficient and scientific manner, by which the "duty" of modern engines has been considerably increased.

The High-pressure Steam Engine.—It is only in recent times that the high-pressure engine, worked by the expansive force of steam alone, has risen into importance, a great stimulus having been given to this description of engine by its universal adoption for railway purposes, owing to its small bulk and light weight, and its requiring no injection water. It was first introduced by Messrs. **Trevithick** and **Vivian** in 1802, and was employed in cases where portability was an object, or where water could not be conveniently had for injection.

The Steam Engine applied to Navigation.

Jonathan Hulls, A.D. 1737.—The first feasible suggestion of the employment of the steam engine to marine purposes appears to be due to Jonathan Hulls, who published a tract in 1737, illustrated by a plate, and entitled "A Description and Draught of a new-invented Machine for carrying Vessels or Ships out of or into any harbour, port, or river, against wind or tide, or in a calm." The plate represents a boat with a paddle-wheel at the stern, propelled by a steam engine, and towing after it a vessel of war. He took out a patent for this invention, but met with so many difficulties, both in the practical application of his views, and in the prejudices of the age, that the project was soon after abandoned, and died at its birth. One reason for this untoward result was doubtless the employment, or at least the suggestion, of Newcomen's engine as the propelling power, before any good method had been invented for changing the unequal rectilinear motion of the atmospheric engine into a uniform rotatory motion.

Marquis de Jouffroy, A.D. 1782.—Hull's suggestion appears to have been first tried in practice by the Marquis de Jouffroy, who constructed a steamboat in 1782, with which he tried numerous experiments on the Soane, at Lyons. They must have proved unsuccessful, however, as they were discontinued about fifteen months afterwards, and the subject was then allowed to drop. The vessel employed was 140 ft. long, 15 ft. wide, and drew 3·2 ft. of water.

Mr. Watt.—Although Mr. Watt's improvements rapidly paved the way for the successful introduction of the steam engine to the purposes of navigation, we do not find that he himself devoted much attention at first to this subject, having chiefly confined his views to perfecting the rotatory engine, and increasing its economy, which was certainly the most important object, and, we may add, at that time the most lucrative for the constructing engineer. Accordingly we find that Watt's rotatory engine was not applied to this use until after the expiry of his patent in 1800.

Miller of Dalswinton, A.D. 1788.—From the years 1785 to 1789 several competitors for the application of steam to navigation appear on the field. In America, there were James Rumsey of Virginia and John Fitch of Philadelphia; in Italy, Serrati; and in Scotland, Mr. Miller of Dalswinton, in Dumfriesshire, whose experiments were by far the most conclusive of any that had yet been made. These were made on Dalswinton Loch in 1788, with a double or twin-boat, fitted with a small engine, made by William Symington, a practical engineer, and propelled by a paddle-wheel placed in the space between the twin-boats. With this they obtained a speed of about three miles an hour. The following year Mr. Miller constructed a larger boat, in which the engine propelled two paddle-wheels, one placed before the other, in the centre between the double boats. The engine in this case was made at the Carron Iron Works, and employed two atmospheric cylinders, each eighteen inches diameter, with which a speed of four to five miles an hour was obtained on the Forth and Clyde Canal, as detailed in the Edinburgh newspapers of the day. This was in the autumn of 1789. After having thus completely established the practicability of the invention, even with the imperfect mechanism employed, Mr. Miller and his colleagues, Symington and Taylor abandoned the further prosecution of the subject, dreading perhaps the expense and opposition which usually falls to the lot of those bold spirits who dare introduce any innovation, however beneficial, on old established customs and prejudices.

Lord Dundas, A.D. 1801.—In the year 1801, Lord Dundas employed Symington to construct a steam-boat, for the purpose of towing barges on the Forth and Clyde Canal. This vessel, called the "Charlotte Dundas," had a rotative engine on Watt's principle, with one horizontal cylinder twenty-two inches diameter and four feet stroke, with connecting rod and crank, which turned a single paddle-wheel situated in a well-hole at the stern of the vessel. It was found to answer its purpose admirably, but the proprietors of the canal objected to its use, on account of the wash of the paddle-wheel, which, they alleged, would injure the banks. The boat was accordingly abandoned in 1802, after an expense of about L.3000 had been incurred by the experiment.

Fulton, A.D. 1803.—It is related that Fulton paid Symington a visit in 1802, and that after minutely inspecting the new steam-boat on the Forth and Clyde Canal he crossed over to France, and laid the invention before the First Consul. His services were fortunately declined, and Fulton then proceeded to America, after having constructed and tried a small steam-boat on the Seine, at Paris, in 1803. His first attempt in America was made on the Hudson river, in 1807, with a steam engine which had been sent out from England by Messrs. Boulton and Watt, the plan of the wheels and gearing being similar to that introduced by Symington six years previously. With this vessel, which was the first American steamer, a speed of about five miles an hour was obtained. Fulton and Livingston then took out a patent for steam-vessels in America, and built several upon a larger scale for carrying goods and passengers, employing Messrs. Boulton and Watt to make the machinery, which was still sent out from England.

Henry Bell, A.D. 1812.—Although steam navigation was thus early introduced on the American rivers, it was not until 1812 that the first regular steam passage-boat, (of tiny dimensions, it is true), made its appearance in this country, on the Clyde. This was the " Comet," built for Mr. Henry Bell, the proprietor of the Helensburgh baths, on the Clyde, and who had been for long a most zealous advocate of steam navigation. The little vessel was forty feet long and ten feet beam, with a steam engine of four horse power on the bell crank principle, the engine being placed on one side the vessel, and the boiler (of wrought iron) on the other. The " Comet " made its first voyage in January, 1812, and continued to ply regularly between Glasgow and Greenock at a speed of about five miles an hour. During this and the succeeding year two other vessels were constructed, of an increased size and power; and in 1814, Mr. Cook, of Glasgow, built a fourth, called the " Glasgow," which, in point of power and efficiency, became the standard at that early period for the construction of river steamers. The kind of engine employed was the *bell crank*, introduced by Boulton and Watt, with a crank on the main shaft, which turned the paddle-wheels on each side of the vessel. The boiler was of wrought iron with internal furnace, and flues surrounded with water.

Boulton and Watt, A.D. 1814.—The marine engines hitherto constructed were single, but in 1814, Messrs. Boulton and Watt applied two condensing engines, connected together by cranks set at right angles on the shaft, for propelling a small steamer on the Clyde.

In 1815, a small vessel with a side lever engine of fourteen horse power, by Cook of Glasgow, made a voyage from Glasgow to Dublin, and thence round the Land's-end to London. It then ran with passengers between London and Margate, with considerable success.

Maudslay, A.D. 1816.—In 1816, Maudslay constructed a pair of combined engines, each fourteen horse power, and applied the power direct to the paddle-wheel shaft, without the intervention of gearing, which was a great improvement. In 1817 Messrs. Boulton and Watt purchased a small steamer, called the " Caledonia," which was built on the Clyde, but had very defective engines. These were removed, and a new pair substituted, of fourteen horse power each, constructed on the side-lever principle, by Mr. James Watt, jun., by whom numerous experiments were made, and some valuable improvements effected.

David Napier, A.D. 1818.—In 1818, Mr. David Napier had the " Rob Roy" built by Denny of Dumbarton, with which he was the first to establish a regular communication between two sea-ports,—namely, Greenock and Belfast. The " Rob Roy" was about 100 tons burthen, and had a single engine of 30 horse power.

Steam from London to Leith, A.D. 1821.— In 1821, Messrs. Boulton and Watt first established a steam communication between London and Leith, before which time great improvements had been made in the construction of the marine engine by Maudslay, Watt, &c. The two largest vessels which had been hitherto constructed were the " James Watt" and the " Soho," with engines of 120 horse power, by Boulton and Watt, and which answered well.

Steam applied to the Royal Navy, A.D. 1819.—In 1819, the " Rob Roy" was transferred from the West coast to the English Channel, to run between Dover and Calais; and about the same time Mr. David Napier built the " Talbot," of 150 tons, and two engines of 30 horse power each, to run between Dublin and Holyhead. In this year the attention of the Admiralty was directed by Lord Melville and Sir George Cockburn to the importance of steam-power for towing men-of-war, and other naval purposes; and the " Comet" was accordingly ordered to be built to the plans and under the superintendence of Mr. Oliver Lang, the master shipwright of Woolwich dockyard. The " Comet" was the first steam-vessel in the Royal Navy; and, we believe, is still in use: she measures 115 feet in length, 21 feet beam, and draws 9 feet of water, being propelled by a pair of engines, of 40 horse power each, by Boulton and Watt.

Aaron Manby, A.D. 1820.— In 1820, a steam-tug was built by Manby for towing barges on the Humber, and in the same year steam-packets were introduced on the post-office station between Holyhead and Howth; and the " Britannia," with oscillating engines, and several other steam-packets, were built by Manby. At this time Messrs. Maudslay and Field applied the expansive action of steam in the cylinder, and also escape valves on the cylinders.

General Steam Navigation Co., A.D. 1825.—In 1825, the General Steam Navigation Company was established by William Jolliffe, and about this period the " Enterprise," 500 tons burthen, built by Gordon, and fitted with a pair of engines of 120 horse power by Maudslay and Field, made the voyage from London to Calcutta, by the Cape of Good Hope. The applicability of steam-vessels, both for river and sea navigation, was now thoroughly established, and steamers rapidly increased in size, power and numbers.

Great Western, A.D. 1838.—The only step which now remained to be accomplished was the crossing of the Atlantic, which was first effected in 1838, by the Great Western. This vessel, which is still considered a model of marine engineering, was designed and built by Paterson, of Bristol, and the engines were constructed by Maudslay and Field. She is 212 feet long, between the perpendiculars; 35 feet, 6 inches beam; and 23 feet 3 inches depth of hold; drawing 15 to 16 feet water when loaded. Her tonnage is 1340 builders' old measurement; and her engines, (on the side-lever construction), are 440 horse power. Her cylinders are 73½ inches diameter, and 7 feet stroke, making 14 or 15 revolutions per minute. She carries about 500 tons of coal, or 15 days' consumption; and her first voyage to New York was made in 13 days, 10 hours.

SECTION IV.

ON THE CONSTRUCTION OF THE STEAM ENGINE.

The steam engine may be primarily divided into two grand classes, namely, the condensing and non-condensing engine; the former employing the power obtained by the condensation of steam in addition to its elastic force, and the latter using the elastic force of the steam alone.

The condensing engine is subdivided into two classes, according as the process of condensation is repeated or not during the ascent as well as the descent of the piston,—hence distinguished by the names of double or single-acting. The double-acting condensing engine is again distinguished according to the various uses to which it is subjected, whether for land or marine purposes. The last of these, namely, the marine engine, we propose considering first; as it will be found to combine within itself most of the appliances and refinements which have been employed for economizing the power of steam; so that, by acquiring a knowledge of this description of engine, we become possessed at the same time of the principles regulating all the other varieties.

Chap. I.—The Marine Engine.

Varieties.—The marine engines constructed and employed in this country act, with very few exceptions, upon the same principle as the double-acting condensing engine, but a large amount of ingenuity has been expended, (with results more or less fortunate), in varying the form and arrangement of the parts to suit the particular circumstances of each case. The old type of the marine engine, which so long stood its ground against innovation, is the side-lever variety, represented in Plates LVI. to LXIII.; and this system has still many advocates for its use, in cases where space and weight are of less consequence than economy in the working and repair of the engines. Side-lever engines began to be first discontinued in the Royal Navy about twelve years ago, the different kinds of direct-acting engines being substituted in their place, in order to save weight and length of engine-room—two very important items in the economy of a war-steamer. A good example of the direct variety is represented in Plates LXXIV. to LXXVII.*; and we also give a view of the principal forms which have been given to this Proteus engine by other makers. (See page 25.)

We now proceed to examine the general action of the marine engine, without reference to any particular type; but the reader should first make himself acquainted with the names and uses of the different parts of those engines, which will be found amply detailed in the descriptions of the Plates above referred to. This will preclude the necessity of our entering into such particulars here, as our space will not admit of such recapitulation.

Steam Pipes and Passages.—The steam pipes communicating between the boiler and the valve casing should be proportioned with a sufficient area to permit a flow of steam to the cylinder at a uniform, and not excessive, velocity. The consequence of making them too contracted is, that the momentum of the steam is destroyed before entering the cylinder, in place of expending itself by impact on the piston; the steam being then what is called "wiredrawn" in the pipes. The usual area given in practice to the steam pipes is one square inch per horse-power, which is amply sufficient for engines of a large size; but in the smaller class of engines this proportion may be increased a little, to allow for the greater comparative consumption of steam per horse-power, as well as the increased comparative surface exposed by the smaller pipe to radiation without and friction within. The rule adopted by Messrs. Boulton and Watt, and some other engineers, is to make the area of the steam-pipe one twenty-fifth of the area of the cylinder.

It is of much importance that the pipes should have as short and direct a route as possible from the boiler to the engine, with few angular bends or changes of direction; all such impediments acting injuriously by checking the momentum of the steam. Where bends are unavoidable, they should be made of as great a radius as convenient. It is also evident that great care should be taken to prevent the loss of heat by radiation, and the consequent condensation of steam in the pipes. They should, therefore, in all cases, be protected by covering them with sheets of hair-felt, wrapped round with spun yarn, and the whole sewn up in canvass and painted. As the least diminution in the temperature of common steam causes a certain amount of condensation, it seems probable that some advantage would be derived by giving the isolated steam an extra dose of heat on leaving the boiler. We do not mean to advocate the use of highly surcharged steam, in which case the consequent diminution of density seems to be an evil more than equivalent to any advantage to be derived from the increase of temperature, but we believe that a slight addition to the sensible heat of the steam would act favourably by preserving its original temperature and preventing condensation in the passages and cylinder, and also by changing into steam the particles of water which are carried over from the boiler in a state of mechanical suspension. Copper is the best material for steam-pipes; the thickness varying from one-eighth to one-quarter of an inch, according to the size. Wrought-iron pipes should be avoided, as they generate scales of rust on the inside, which, becoming detached, are blown by the steam into the valves and cylinder, where they do much mischief by scratching and cutting the surfaces. It must be admitted, however, that copper steam-pipes exert a galvanic influence upon the cast-iron of the valve casing, which suffers corrosion, as being the more oxidable metal. The steam-pipes are subject to considerable wear from the friction of the steam passing through them. When a straight pipe forms the connection with the boiler, an expansion or "fawcett" joint must be provided to allow for the expansion of the metal by the heat of the steam, but when an elbow occurs in the length of pipe, which is usually the case, the expansion joint may be dispensed with.

The Throttle Valve.—We may suppose the steam to have now arrived at the throttle valve, which is a brass disc turning on an axis, and fixed by hand in such a position as to pass the amount of steam required for keeping the engine at the desired speed. The legitimate use of the throttle valve in the marine engine is for controlling the speed of the engines for any temporary occasion, such as in passing through a crowded river, or before stopping at a pier, &c.; but it should seldom or never be employed for working the engines expansively, at a permanent reduction of speed.

Expansion Valves.—This object is much better effected by employing a distinct valve for the purpose, called the expansion valve, which should be fitted to all sea-going steamers. The expansion valve usually derives its motion from the crank shaft of the engine, as represented in Plates LVI. and LXXIV., the valve spindle being connected, by a series of levers, with a small brass roller kept constantly pressed against the circumference of a cast-iron cam, by means of a counterbalance.

Fig. 1.

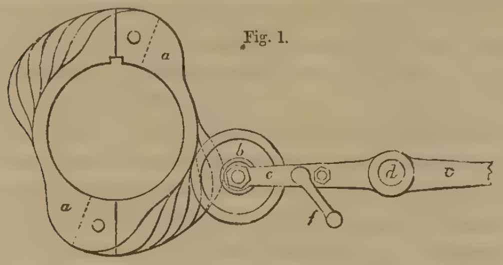


Fig. 2.

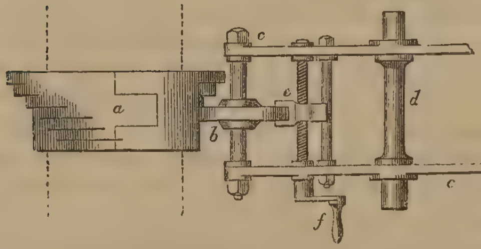


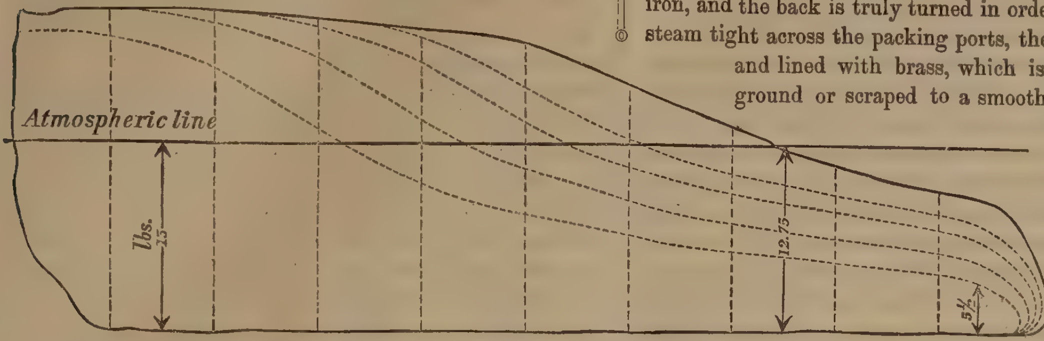
Fig. 1 is an elevation, and Fig. 2 a plan of the apparatus, as fitted in H.M. St. Frigate Cormorant. *a*, the cam, having seven steps, and so constructed as to admit of being fixed upon the crank shaft with the utmost facility; *b*, the roller, capable of moving longitudinally upon its axis in the extremity of the double lever *c*, for the purpose of being brought into contact with either of the steps of the cam, according to the amount of lift required for the valve to give the desired degree of expansion; *d*, the centre of motion of the lever; *e*, a nut working into a screw situated parallel to the axis of the roller, and so formed as to embrace the edge of the roller, thereby causing it to move longitudinally as the screw is turned by means of the handle *f*. By an arrangement of this kind the steam is cut off in the most advantageous manner at any given part of the stroke.

The valve itself is represented in detail in Plate LXXVII,* Figs. 4 and 5, the latter figure showing the sliding portion of the valve to which the spindle is attached. It is of the description called the Cornish double-beat or equilibrium valve, and is formed of gun metal carefully turned and bored. This kind of valve has the advantage of being opened and shut with great facility, as, from its cylindrical form, the pressure of the surrounding steam has no tendency to jam it against the seat, which is the objection to which all flat or plate valves are subject. Also by a slight rise of this valve a very large opening is obtained for the steam. Other ingenious kinds of expansive apparatus have been applied to the marine engine, such as a flat disc made to turn round on an axis within the steam-pipe, which is the usual American method, and more recently a modification of Stephenson's simple and elegant link motion for locomotives. By adopting the latter contrivance, the length of stroke of the cylinder slide valves is varied at pleasure, so that they themselves act the part of expansion valves, but as their motion is derived from an eccentric, it has a different character from that produced by the cam in the former instance; the admission and exclusion of the steam being now performed more gradually and, in our opinion, with less effect.

The link motion is explained at Plate III., Figs. 8 and 9, where *c* represents a curved link along which the head of the valve rod, D, is free to slide in a double groove from one end to the other. Two eccentric rods, B, B, to which a contrary motion is communicated by their respective eccentrics on the crank shaft, grasp the extremities of the link; one for going ahead, and the other for going astern or reversing the engines. Now, when the valve-rod is in the position represented in the Plate, namely, in the centre between the two eccentrics, no motion will be communicated to the valve, as the centre of oscillation of the link passes through the connecting-pin of the valve-rod without imparting to it any horizontal motion; but if we slide the rod D nearer either end of the link, it is apparent that a linear motion will be communicated to it, greater or less in proportion as the distance between the centre of the valve-rod and the centre of the link is increased or diminished. Thus when D has been moved to the extreme limit at either end of the link, the engine will be working with full steam, either ahead or astern according to which eccentric imparts the motion to the valve-rod, and any intermediate degree of expansion is obtained simply by bringing the rod nearer the centre of the link. As already remarked, we think the action of this apparatus too gradual in cutting off the steam, believing that the most economical method of working expansively is to admit the full pressure of steam suddenly into the cylinder, so that it may act by its momentum or impact in quickly destroying the *vis inertiæ* of the piston, and giving it a good start forward; the flow of steam being then continued uninterruptedly until the completion of the desired fraction of the stroke, when the expansion valve should be suddenly closed and all further supply of steam cut off. The link motion has been applied to the valves of the marine engine by Messrs. Maudslay and Messrs. Rennie. An additional expense is incurred by the use of two eccentrics in place of one, but this is compensated by the increased facility in working the engines of this motion, which does not require the eccentric rod to be put out of gear, but merely to be moved into the middle of the link in order to stop the engine.

The indicator diagram, Fig. 3, is taken from an engine by Rennie, and exhibits the different degrees of expansive action obtained by his patent link motion as applied to the "Bull dog," &c.

Fig. 3.



The following rule will be found useful for calculating approximately the mean pressure of steam exerted on the piston during the stroke, while working expansively. The only correct method of arriving at the power of an engine is by the use of the indicator (explained in another portion of this Work),* but as this instrument cannot be applied to an *ideal* engine the rule and table here given will be of use for estimating the probable power of the engine beforehand, for different degrees of expansion.

Rule.—Divide the length of stroke by the distance the piston moves before the steam is cut off, and the quotient will express the relative expansion it undergoes. Take from the annexed table the multiplier corresponding to this number, and multiply it by the full pressure of steam per square inch on entering the cylinder; the product being the mean pressure per square inch.

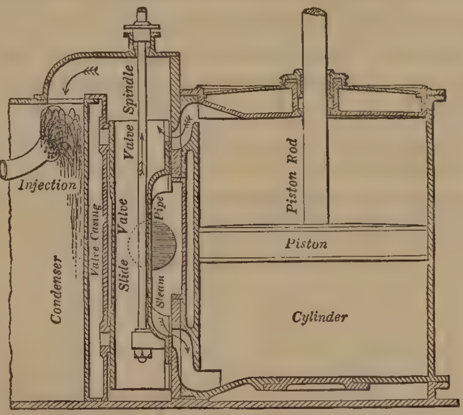
If the number corresponding to the relative expansion found as above, falls between two of the printed numbers, we must assume a new decimal for the multiplier in proportion to this difference.

TABLE OF MULTIPLIERS.

Relative Expansion.	Multiplier.	Relative Expansion.	Multiplier.	Relative Expansion.	Multiplier.	Relative Expansion.	Multiplie
1	1·0000	2	·8466	3	·6995	4	·5966
1 1	·9957	2·1	·8295	3·1	·6875	4·1	·5880
1·2	·9853	2·2	·8129	3·2	·6760	4·2	·5798
1 3	·9710	2·3	·7969	3·3	·6648	4·3	·5718
1·4	·9546	2·4	·7814	3·4	·6540	4·4	·5640
1·5	·9370	2 5	·7665	3·5	·6436	4·5	·5564
1·6	·9188	2·6	·7521	3·6	·6336	4·6	·5491
1·7	·9004	2 7	·7382	3·7	·6239	4·7	·5420
1·8	·8821	2 8	·7249	3·8	6145	4·8	·5351
1·9	·8641	2 9	·7120	3·9	·6054	4·9	·5284
2	·8466	3	·6995	4	·5966	5	·5219

Cylinder Slide-Valves.—The cylinder slide-valves which have been usually applied to the marine engine belong to one of two kinds called, from their sectional shape, the short D slide, and the long D slide. The first of these is represented in Plates LXI., LXII., and the other is shown by the wood-cut, Fig. 4, and also in Plates XIX., XX., as applied to a stationary condensing engine. The principle of both is exactly alike, and the only difference in their operation is that with the short slide the eduction takes place through two separate passages to the condenser, one for the upper port and the other for the lower; whilst the long D valve, being hollow, and passing steam tight through the space occupied by the steam in the valve casing, itself forms the eduction passage. The valve is usually of cast-iron, and the back is truly turned in order that it may move steam tight across the packing ports, the face being planed and lined with brass, which is again planed and ground or scraped to a smooth and true surface. Until recently, the valve facings were always ground with emery, but this substance is found to incorporate itself, during the operation, with the surface metal, and thus to cause unequal wear upon the cylinder face when the valve is set to work. It is now generally disused except in a finely pulverized state for giving a last finish to the work; steel scrapers being substituted in its place. The back of the valve is usually packed with hemp pressed close by metal packing rings and set screws, which may be screwed up without taking off the covers of the packing ports in the valve casing.

Fig. 4.



The manner of setting the valves, so as to admit and shut out the steam from the cylinder at the proper time, is a matter of the highest importance. We have already discussed the propriety of shutting off the steam a little before the end of the stroke, by which means the piston is brought gradually to rest without jarring the engine, independently of the advantage derived from expanding the steam.

It is also of much consequence that the communication with the condenser should be opened, on the opposite side of the piston, a little before it reaches the end, in order that time may be allowed for the steam to rush through the eduction passages, and produce a vacuum in the cylinder before the commencement of the return stroke. This also assists in checking the injurious velocity of the piston at the end of the stroke. It is also the usual practice to adjust the valves, so that the steam-port may be a little open when the piston arrives at the top or bottom of its stroke; this space (varying from $\frac{1}{8}$ to $\frac{1}{4}$ inch) being called the "lead" of the valve. The double operation of cutting off the steam and opening the eduction port before the end of the stroke is performed by causing the slide-valve to "travel" past the ports for a certain space in each direction, such space being called the "lap" or "cover" of the valves. By allowing lap to the valve, we also modify, in the most advantageous manner, the motion it receives from the eccentric; which has been accused, and with apparent justice, of wire-drawing or attenuating the steam at the commencement of the stroke; but by giving the valves a longer stroke, and causing them to travel past the ports, the motion communicated by the eccentric will have become more rapid before the port opens to steam.

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* See Plate CXXII. and its description.

It is apparent that the lap of the slide-valve offers a simple method of working the steam expansively; and, indeed, it is usually adopted with this end in view. The degree of expansion which can be advantageously given by the slide-valve is limited, as it necessarily follows that a valve with much lap on the steam side will close the eduction-port considerably before the piston has reached the end of the stroke; opening at the same time the eduction passage to the steam on the expanding side before it has finished its work.

The annexed table shows the operation of the slide-valve for an engine of 150 horse power in the Government service; the valve-face and cylinder-ports being represented by the wood cut, Fig. 4:—

Length of stroke of piston, ...63 inches.
Stroke of the valve,......................................$13\frac{1}{4}$ —

Port open to Eduction.	Stroke of Piston.	Port open to Steam.	Port open to Eduction.	Stroke of Piston.	Port open to Steam.
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
$2\frac{5}{8}$	On centre	$\frac{1}{8}$	$4\frac{3}{4}$	45	$2\frac{1}{4}$
$4\frac{13}{16}$	3	$2\frac{5}{16}$	$4\frac{1}{4}$	48	$1\frac{3}{4}$
$5\frac{1}{2}$	6	3	$3\frac{5}{8}$	51	$1\frac{1}{8}$
$5\frac{7}{8}$	9	$3\frac{3}{8}$	$2\frac{7}{8}$	54	$\frac{3}{8}$
$6\frac{3}{16}$	12	$3\frac{11}{16}$	$2\frac{1}{2}$	55	Shut to steam
$6\frac{3}{8}$	15	$3\frac{7}{8}$	$1\frac{3}{4}$	57	$\frac{3}{4}$ covered
bare $6\frac{1}{2}$	18	4 bare	$\frac{1}{2}$	60	2 covered
full $6\frac{1}{2}$	21	4 full	Shut to Eduction & covered $\frac{3}{8}$	$61\frac{1}{4}$	Eduction begins on opposite side of port.
full $6\frac{1}{2}$	24	4 full			
bare $6\frac{1}{2}$	27	4 bare			
$6\frac{5}{16}$	30	$3\frac{13}{16}$	$\frac{7}{16}$	$61\frac{3}{8}$	$\frac{1}{16}$ Open to Eduction
$6\frac{1}{8}$	33	$3\frac{5}{8}$			
$5\frac{7}{8}$	36	$3\frac{3}{8}$	$2\frac{7}{8}$	63	$2\frac{1}{2}$ Open to Eduction
$5\frac{3}{4}$	39	$3\frac{1}{8}$			
$5\frac{1}{4}$	42	$2\frac{3}{4}$			

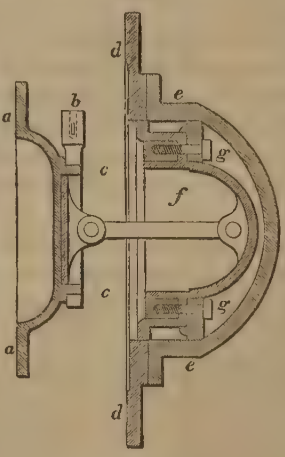
It will be observed that in this case the eduction begins on the steam side of the piston, at $1\frac{3}{4}$ inches before the end of the stroke. The steam-port is quite shut to steam at 8 inches before the end of the stroke, the steam beginning to be shut off a little before half-stroke. The lead to steam, when the crank is on the centre, is $\frac{1}{8}$ inch, and to eduction $2\frac{5}{8}$ inches; the uncondensed vapour under the piston being shut up for $2\frac{1}{2}$ inches at the end of the stroke, before the port opens to steam.

If we wish to find how much lap must be given on the steam side, in order to cut off the steam at any given part of the stroke, the rule will be as follows:—"From the length of stroke of the piston, subtract that fraction of the stroke which is to be made before the steam is cut off. Divide the remainder by the whole length of stroke, and extract the square root of the quotient. Multiply the root thus found by half the length of the stroke of the valve, and from the product take half the lead; the remainder being the required lap or cover."

Starting Gear.—As the valve must first be set agoing by hand, before the engine can make a stroke and continue the motion, an apparatus for this purpose must be provided. In the case of large engines, starting the valves is often a work of much difficulty, and a correspondingly powerful apparatus is required. The usual way with large engines is to disconnect the eccentric rod, and connect, at the same time, the head of the valve spindle with the axis of a large starting wheel by some such combination of levers and gearing as represented at Plate LXXIV., LXXV., by which means the power of a man may be easily multiplied from ten to fifteen times. After the piston and other moving parts of the engine have acquired such momentum as will continue the motion, the starting gear is suddenly disengaged, and the end of the eccentric rod is dropped into gear, which then works the valves by virtue of the acquired momentum of the machine. In some few instances, as in the *British Queen*, a small engine is employed for this purpose, but by proper management, the valves, even of the largest engines, may now be worked by hand.

In the case of oscillating engines, the valves employed are short box valves, similar to those used for locomotives, as represented at Fig. 5. As these valves are difficult to work on account of the great pressure of the steam on the back, a contrivance such as that shewn in the woodcut is sometimes adopted for relieving them of a part of this load; *a* is the sliding valve formed of brass; *b*, the valve spindle; *c*, the steam space at the back of the valve; *d*, the valve casing; *e*, a hemispherical box of cast-iron fitted on the outside of the casing opposite the slide valve; *f*, a hollow piston working in the box *e*, and connected by a jointed rod with the back of the slide valve. The space *g*, round the back of the piston, is brought into communication with the condenser, by means of a small connecting pipe, so that we have now steam on the front of the piston and vacuum behind it. The consequence is, that the pressure of steam on the piston counterbalances and relieves the pressure on the sliding valve, by tending to pull it off the face of the cylinder through the jointed rod.

Fig. 5.



Steam Ports.—Theoretically speaking, the steam ports should be made as large as possible, but in practice a usual and convenient proportion is to give an area of one square inch per horse power to each port, or to make each from $\frac{1}{20}$ to $\frac{1}{24}$ part of the area of the cylinder, the smaller cylinders having proportionably larger ports. The stroke of the valve, and the lap to be allowed, will determine the breadth of the ports. For an 80 inch cylinder we would recommend a stroke to the valves of about 15 inches; for a 65 inch cylinder, $13\frac{1}{2}$ inches; for a 40 inch cylinder, 10 inches; for a 30 inch cylinder, 8 inches, &c. It is a usual practice to allow such lead to the valves as will give an opening to steam of $\frac{1}{16}$ of a square inch, per nominal horse power, (or 1 square inch per 16 horse power), when the crank is on the centre; and it may be taken as a safe rule that, in all cases, expansion may be advantageously given by the slide valve after $\frac{5}{8}$ of the stroke of the piston.

Slide Valve Spindles.—The slide valve spindles may be made $\frac{1}{24}$ of the diameter of the cylinder.

Eccentric.—The eccentric is invariably used for working the valves of marine engines. Its action is precisely similar to that of the crank, although attended with much greater friction, but it will be seen that the crank is quite inapplicable in this case, owing to the space it requires, and its weakening effect upon the shaft. The stroke or "throw" of the eccentric is, of course, equal to twice the eccentricity of the inner circle keyed on the paddle shaft. The outer circle, which communicates the motion to the valve lever or "weigh shaft," is either formed entirely of brass, or fitted with a lining of that metal, in order to reduce the friction between it and the periphery of the cast-iron eccentric block.

Blow-through and Snifting Valves.—Before the engine can be started, it is necessary that the air should first be expelled from the cylinder, condenser and air-pump, and its place supplied with steam, in order to produce a vacuum by its condensation. Hence, a valve, which is called from its office the "blow-through valve," must be provided to open a temporary communication between the steam in the valve casing and the condenser, by which a rush of steam may be caused to pass through the internal parts of the engine. This operation is continued until the steam begins to issue hot and transparent from another valve called the "snifting valve," situated at the opposite point from where the steam enters. The blow-through valve is seen at *b*, Plates XIX., XX., and at x', in Plates LXXIV., LXXV., forming a junction between the interior of the valve casing and the condenser, and the snifting valve is conspicuous at u', in Plate LX. These valves are, of course, closed as soon as the engine is in regular operation.

Eduction.—The area of the eduction passages should be the same as those for steam.

Proportion of Diameter to Stroke of Cylinder.—The most advantageous proportion between the diameter of the cylinder and the stroke of the piston is a subject which can seldom enter into the calculations of the marine engineer, who must be content to accommodate the length of stroke to the other circumstances of the case. Thus the only rule we can give for marine engines is, to make the stroke as long as can be conveniently done. In land engines, which are unfettered by such considerations as the height of the crank shaft, the centre of gravity, the saving of weight and space, &c., the best proportion appears to be, both from theory and practice, to make the stroke equal to twice the diameter of the cylinder; but, in these days of direct acting engines, the marine engineer may think himself fortunate if he can obtain a stroke equal to the diameter.

Direct Acting Engines.—The unavoidable shortness of the stroke and connecting rod, in the majority of the direct acting engines, are certainly defects, and become sensible in practice by an increased wear and tear of brasses and packings, and a greater consumption of tallow and oil, as compared with the side-lever engine. Several of the direct acting varieties, it is true, are not so confined in the length of stroke; as, for instance, the steeple engine, which is a favourite on the Clyde, and which would be an excellent arrangement, but for the great height of its centre of gravity and the large scuttle it requires in the deck, which make it somewhat inapplicable to sea-going vessels. Maudslay's double cylinder variety also makes an excellent engine, and may have a good long stroke, but it is heavy and expensive for small powers, and takes up a good deal of room as compared with other direct engines, although it appears to be most unexceptionable and efficient for engines of a large class. Of all the direct kinds, however, we prefer none to the oscillating engine of Mr. Penn, to whom it owes so much of its elegant simplicity, and present perfection of workmanship and arrangement. We believe this construction has now been fully proved to be as applicable to engines of 400 and 500 horses power as to the small steamers on the Thames, where it has long been a favourite, and we think it threatens shortly to supersede most other varieties for driving the screw propeller.

We would not have it inferred from the above remarks that we do not appreciate the advantages of the direct acting engine, which, on the contrary, we believe to have been a most valuable acquisition to our steam marine, more especially the Royal Navy; but what we allege is, that it has been proved to be inferior, as a machine, to the side-lever engine, both as regards economy of power and wear. As, however, the side-lever engine is about $\frac{1}{3}$ heavier, and occupies perhaps $\frac{1}{3}$ more space in the vessel than its rival, it is not surprising that it has been in so many cases superseded.

Calculation of Nominal Horse Power.—The arbitrary calculation which, by common consent, has been designated the horse power of an engine, is made as follows, viz.:—

$$\frac{\text{Area of cylinder} \times \text{effective pressure} \times \text{speed of piston}}{33{,}000} = \text{H.P.}$$

The area must be taken in square inches, the effective pressure being assumed at 7 to $7\frac{1}{2}$ lbs. per square inch of piston, (usually the latter), and the speed reckoned in the number of feet through which the piston travels, or is expected to travel, per minute. The divisor, namely 33,000 lbs., was James Watt's standard of a horse power, when supposed to be raised one foot high in a minute. In tendering for engines for the government service by the horse power, the effective pressure is taken at 7 lbs. only, and the speed of the piston is supposed to vary with the stroke according to the following table:—

Stroke.		Speed.	Stroke.		Speed.
feet.	in.	Feet per Min.	feet.	in.	Feet per Min.
4	0	196	6	6	226
4	6	204	7	0	231
5	0	210	7	6	236
5	6	216	8	0	240
6	0	222			

It will be at once seen that the horse power thus calculated is irrespective either of the actual pressure of the steam used, of the perfection of the vacuum in the condenser, or of the amount of friction arising from good or bad workmanship—hence this rule is no guide to the real power of an engine. It serves well, however, as a conventional standard for comparing the size of different engines, of whose actual performance we have no opportunity of judging.

It is usual among engineers of the present day to abuse this rule, as being absurd and useless; but we do not see what benefit would be derived by the use of a more practical rule applied to an ideal engine, as we rather think it must be more satisfactory to the purchaser of a steam engine to know that he is receiving for his money a definite size of engine, than to rest his hopes for a sufficiency of power on the manufacturer's mere expectations and promises of large performance. As the case stands, we have two distinct values of horse power, which no one, in the least conversant with the steam engine, ever confounds; the one being fixed and nominal, by which engines are bought and sold; and the other fluctuating, but real, as shown by the indicator.

Cylinder.—But to return to the cylinder. The finished thickness of metal varies from about 1 inch for small powers to $1\frac{1}{2}$ inch for the largest class of engines. In castings of this nature it is not sufficient that we proportion the thickness of metal simply to the strain or pressure; we must allow a sufficient thickness to ensure the perfect running of the metal into every part of the mould before it begins to consolidate. Also, it is better, in the case of a cylinder, to allow a little extra thickness at first as a provision for the re-boring of the cylinder after it has been some time at work. The casting should always be made in loam, or dry sand, with plenty of blackening brushed over the surface, and the metal used is generally made up of several different kinds mixed together, which has the effect of making the casting hard and tough. The following is the mixture for cylinder castings, adopted by Mr. Fairbairn, who has devoted so much of his attention to the properties of cast-iron, viz.:—

Pentwyn iron,	No. 2,	20 parts.
Coed Talon,	— 3,	20 —
Apedale,	— 3,	20 —
Clyde,	— 3,	12 —
Butterly,	— 3,	6 —
Scraps,		5 —

Cylinders of a large diameter should be bored, if possible, in the upright position, as they take a slightly oval form by their own weight when laid horizontally. This defect is inappreciable in small cylinders, but becomes apparent when they reach a diameter of about 60 inches. The boring of a large cylinder is a very tedious operation, in consequence of the slow speed at which the cutting tool must travel, and for a 250 horse cylinder the tool performs a spiral journey of about ten miles round the interior of the cylinder before the operation is complete. A cylinder $65\frac{1}{2}$ inches diameter and 80 inches long took the following time to bore, viz., the first or "roughing" cut took 116·36 hours; the second or finishing cut 160 hours; the travel of the tool per hour being for the roughing cut about $\frac{11}{16}$ inch, and for the finishing cut about $\frac{1}{2}$ inch. The motion must be kept up uniformly by day and night during the whole time of the finishing cut, otherwise the work may very probably be untrue.

Metal Joints.—The cylinder is usually attached to the base-plate by what is termed a *rust joint*, formed by a cement composed of one ounce of sal ammoniac in powder, about eighteen ounces of fine cast-iron borings, and half an ounce of flowers of sulphur. After being well mixed together, this compound is wetted with a little water, and in a few hours time it is ready for use, being driven into the open joint, from the outside, by a square-edged caulking iron. The skin of the cast-iron must be removed before attempting to make the joint, and all grease carefully washed off with sulphuric acid. Where the joint is likely to be broken for the purpose of examining or repairing the engine, this cement should on no account be employed, as it soon sets so hard as to form almost an integral part of the iron surfaces it unites. Such joints should therefore be made with sheet lead or wire gauze plentifully smeared with a thick paste of red lead and linseed oil, the iron surfaces having been previously chipped and filed true to each other.

Clothing the Cylinder.—The cylinder and valve casings should be invariably clothed with some non-conducting substance, more especially in these latter times of tubular boilers, high steam and expansive working. The substances usually employed for this purpose are hair felt, laid on about two inches thick, and covered with dry fir timber bound round the cylinder by iron or brass hoops. The loss of power from the cooling of the steam within a naked cylinder has been variously estimated at one or two per cent., but in the case of steam of 8 or 10 lbs. pressure we believe it exceeds this amount. It is a well known fact that in Cornwall any attempt to discontinue the use of the steam jacket round the cylinder has been attended with a palpable loss of power, although the radiation of heat still seemed to be effectually prevented by the use of the best non-conducting substances. This shows of how much consequence it is to preserve the internal heat of the cylinder, and would appear to point out the applicability for this purpose of steam slightly surcharged with heat, by which means, also, as we have before remarked, the particles of water held in mechanical suspension by the steam would become vaporized.

Clearance.—The "clearance" of the piston at top and bottom of stroke should of course be as little as may be consistent with safety, and is usually made from $\frac{1}{2}$ to $\frac{5}{8}$ or $\frac{3}{4}$ of an inch. It will be seen that the steam which occupies this space, as well as that contained in the steam ports, is in a measure lost, but in working expansively the clearance steam has its value.

Escape, or Priming-Valves, on Cylinders.—Escape-valves are now always fitted by respectable makers at the top and bottom of the cylinder, to allow egress to the water which may have collected there, without any danger being incurred from the shock of the piston against the incompressible fluid. This water collects either from the condensation of steam within the cylinder, or it may be carried over with the steam from the boiler; or again, it may overflow through the valves from the condenser. When the water in the boiler is too violently agitated, it sometimes boils over into the steam-pipe, when the boiler is said to "prime;" and the cylinder escape-valves, having to eject this water, are hence frequently called the "priming-valves." Besides the water from priming, a certain amount is carried over with the steam into the cylinder, in a state of mechanical suspension.

The area of each cylinder escape-valve may be found in square inches, by multiplying the horse power of the cylinder by ·14. They are made of brass, and loaded either by a weight or a spring to somewhat above the pressure in the cylinder; and should be fitted with hoods to carry off the ejected water into the bilge, and prevent its being thrown about the engine-room. An example of their application is seen at *g′ g′*, Plates LX., LXI.; as also in Plates LXXVI., LXXVII.

Piston.—The Piston, which is so important a feature in the working of the steam engine, has been subjected by manufacturers to an almost infinite variety of form. Of these some of the best will be found detailed in our Plates of land, marine, and locomotive engines.

The packing of the piston, which has been the chief object of improvement, was originally formed of hemp; but as this substance, after being some time in use, becomes hard and inelastic, and ceases to form a steam-tight joint, besides having a tendency to wear the cylinder unequally, it has now been almost universally abandoned for metallic packing. This is employed in the form of cast iron rings, either possessed of elasticity in themselves, or deriving it from springs placed behind them. When elastic rings are used, they are cast of a diameter a little larger than the cylinder they are intended to fill; and a small segment of the circle being cut out, the ends are sprung together, and the ring is thus introduced into the cylinder. Two or more of such rings must be used together, in order that the open joint at the ends of the one ring may be covered by the other, and prevent the passage of steam which would otherwise occur at the place where the ring was cut. They are sometimes cast eccentric, being thick at one end of a diameter and thin at the other, the segment being cut out of the thin part; by which it is intended that the elasticity shall be more uniform throughout when the ends are sprung together. In such a case four rings must be employed; the one set being placed behind the other, in such a position that the thick part of the one ring comes behind the thin part of the other. This arrangement makes a very good piston for a while, but the metal rings unfortunately lose their elasticity after being some time in use, so that they must either be taken out and have their elasticity renewed by hammering, and therefore elongating, their internal surface, or else steel springs must be applied at the back to press them out against the cylinder. When the metal rings depend upon springs for their elasticity, they are cut into segments to allow the springs a greater freedom of action.

As an additional security against leakage, hemp is sometimes introduced into the space behind the packing-rings, being compressed by means of a "junk-ring," which forms part of the top of the piston. The body of the piston is usually formed deeper at the centre than at the circumference, and the "boss" or hole for the end of the piston-rod is made tapering outwards, at the rate of about one inch in twelve of depth. The end of the rod is secured either by a cutter or screw, or by both conjointly, in the manner shewn at Fig. 2, Plate LXXVI., which is a very safe arrangement. The piston is supplied with melted tallow through a small grease-cock in the cylinder-cover, advantage being taken of the vacuum in the up-stroke to suck in the tallow, which runs down the curved surface of the piston to the outer edge. The piston is also lubricated by the tallow which leaks through the gland of the cylinder-cover. We may observe, that the aperture for the grease-cock usually serves also for connecting the indicator with the interior of the cylinder in applying it to determine the character of the up-stroke.

Piston-Rod.—The piston-rod is formed of the best faggoted wrought iron, and is frequently steeled or case-hardened on the outside, either by heating with charcoal in the usual way for forming steel, or by bringing the rod to a dull red heat, rubbing it over with prussiate of potash, returning it to the fire for a few minutes, and then quenching with water. The usual practice in London is to make the piston-rod one-tenth part of the diameter of the cylinder; and if for oscillating engines, of steel of the same diameter.

Condenser.—The condenser of an engine has usually the capacity of half the steam cylinder, and it should not be less than this if possible. Some makers adopt the rule of one-third of a cubic foot per horse power, and others make its capacity in proportion to that of the cylinder, as the numbers 8 : 14. The size of the condenser should, however, be proportioned to the temperature, or density, of the steam which it is proposed to use, as plenty of space should be allowed for high steam to expand into low-pressure steam before being condensed, if this has not been previously accomplished by expansive action within the cylinder.

Injection.—The area of the injection-valve and orifice, for a marine engine, should be about one square inch for every ten horse power. This is an ample allowance in all cases, and more than necessary when the injection water has a temperature of 52°, which is the average for our seas; but in tropical latitudes it will not be found too much. The average temperature of the sea in the Mediterranean is 65°; at the twentieth degree of latitude, about 75°; and at the equator, about 82° Fahr. The supply is regulated at will by the opening given to the valve or cock, as shown by an index and plate connected with the spindle. It is frequently a sluice-valve, as represented at *w*, in Plate LXXIV., and is formed of brass. It is better in the case of large engines to have two injection cocks; one, (called the *sea-cock*,) on the side of the vessel, for greater security in case of injury to the pipe; and the other on the condenser. The kind of valve represented by Figs. 6 and 7, called Kingston's valve,

Fig. 6.

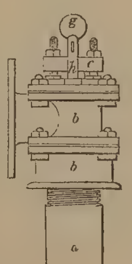
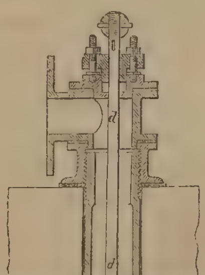


Fig. 7.



Scale, 1 inch to a foot.

is probably the best for forming such communications with the sea, especially for wooden vessels. It is formed entirely of brass or gun-metal, and the injection pipe of copper.

Fig. 6 is an external, and Fig. 7 a sectional, elevation of Kingston's valve, as adapted to fulfil the purpose of a sea-cock: *a*, the pipe, of gun-metal, passing through the ship's side or bottom; *b*, the cover, or upper part of the pipe, screwed to the former, and having a branch for connecting with the injection pipe; *c*, a water-tight stuffing-box, through which the spindle, *d*, *d*, of the valve works; *e*, the valve itself, screwed firmly to the spindle, which is guided at its outer extremity by a socket formed upon a guard or grating, *f*, *f*, fixed into the mouth of the pipe, for the purpose of excluding weeds, or other foreign bodies; *g*, the handle of the valve; *h*, a copper standard screwed into the cover (one on each side of the stuffing-box), through which, as well as through the valve-spindle, *d*, *d*, are formed mortices or slots for the reception of a key, for the purpose of maintaining the valve in its place when fully opened; when this key is withdrawn, the pressure of the water outside the vessel instantly shuts the valve.

Mr. Watt's practice was to allow a pint, or twenty-nine cubic inches, of injection water for every cubic inch of water converted into steam. The condensation will be the more complete in proportion to the coldness and quantity of the injection, the manner in which it is brought into contact with the steam, and the temperature of the steam itself on entering the condenser. The resulting temperature which is usually aimed at by engineers for the condenser, is about 100° to 120°. The limit to a more perfect degree of condensation is imposed in practice by the necessity of increasing the size of the air-pump to withdraw the additional amount of injection water required, which tends both to diminish the effect and increase the expense of the engine; so it is evident that there is a point beyond which it becomes injurious to push the process of condensation by injection. Hence it is not desirable to aim at a temperature below 100° or 110°, with which a vacuum of 27½ or 28 inches of mercury is obtained by a good engine.

To preserve this temperature in the condenser requires, theoretically, about twenty to twenty-two times the weight of the steam, in water of 52°. According to Dr. Ure's experiments, the following table represents the force of the vapour of water within the condenser, in inches of mercury, for different temperatures, viz.:—

Fahr.		Inches.
60°	produces a pressure of	0·516
70		0·726
80		1·010
90		1·360
100		1·860
110		2·456
120		3·300
130		4·366
140		5·770
150		7·530
160		9 600
170		12·050
180		15·160

High-pressure steam requires, of course, more injection water in proportion as its temperature is higher. The manner in which the jet of steam is brought into contact with the condensing water is a matter of much importance. The object is, that the surplus heat of the steam may be transferred to the water in as short a time as possible, since the perfection of the vacuum depends very much upon the rapidity of the process of condensation. Now, although water has an advantage over most substances in possessing a very large capacity for heat, it is by no means a good conductor, and must therefore be presented to the steam in a state of extreme subdivision. This is effected either by attaching a *rose* to the orifice of the injection pipe, by which the water is separated into a shower, or by causing it to strike forcibly against the top of the condenser. The velocity with which the stream of water enters the condenser varies as the square root of the pressure; and will be about forty feet per second for the pressure of the atmosphere against a vacuum. The sea-injection-cock should draw from about midway between the surface of the water and the bottom of the keel, so as neither to draw impurities from the surface nor be liable to be choked with sand or mud from the bottom.

A barometer is attached to the condenser to show the vacuum; it is constructed like a common barometer, except that the top of the tube communicates by a small pipe and cock with the interior of the condenser, the partial vacuum of which thus takes the place of the Torricellian vacuum. In the case of sea-going steamers it is a usual and highly commendable practice to lead an injection pipe from the bilge of the vessel into the condenser, so that, should any leakage occur, the sea-injection-cock may be shut and the engines supplied with injection from this source, the mouth of the pipe being, of course, carefully guarded and kept clean, in case of sudden emergency. This duplicate condensing apparatus is a *sine quâ non* in the Government service.

Condensation by Surface.—Many attempts have been made to condense the steam by contact with cold metallic surfaces in place of water. In the case of marine engines employed in salt water, this is a most desirable object, but unfortunately the principle of surface condensation has hitherto always proved inefficient. This arises chiefly from the process of condensation being too protracted, owing to the difficulty experienced in presenting a sufficiently large cooling surface to the steam. In Hall's condensers, which have received a very fair trial, but which are now rarely met with, the steam is passed through many miles of little pipes enclosed in a cistern of cold water, which is constantly renewed from the sea, by a force-pump worked by the engine. The boiler is fed with the condensed steam, the loss arising from leakage and the action of the air-pump being compensated by the use of a small apparatus for distilling sea-water. All this additional machinery adds to the expense and intricacy of the engine, besides being objectionable from the extra weight and space required for the large cold water cistern containing the tubes, and the latter are, besides, liable to be incrusted, or choked up entirely, with deposit from the water.[*] In such an arrangement the air-pump is much reduced in size, as it has now no injection-water to remove, but only the air and uncondensed vapour, the whole of the condensed steam being returned by the feed-pump into the boiler. Other modes of condensation by surface, as it is called, have been attempted with more or less success, the most ingenious, perhaps, being that of Mr. David Napier, who made the bottom of the iron boat itself the condensing surface.

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[*] Hall's condensers on board the British Queen possess a condensing surface of about 18 square feet per horse power, and require 7 gallons of water per horse power per minute, to force which requires about 12 horse power, or a consumption of 1 cwt. of coal per hour. The weight of the condensing apparatus is 40 tons, which is equal to a displacement of 3 inches of the vessel, and will cause a loss of perhaps 5 per cent in speed and consumption of fuel. Her average

The use of sea-water for condensation entails upon marine engines a long catalogue of ills which may be enumerated as follows. The salt water, boiling at a temperature of about 220°, in place of 212°, requires more heat for raising steam than fresh water; and the danger to which the boiler would be subjected by an undue degree of saturation, and a consequent deposition of salt, entails the necessity of frequent "blowing-off" into the sea, by which a considerable quantity of heat is wasted. The specific gravity of the salt water in the boiler, at a medium degree of saturation, will be about one-tenth greater than that of fresh water, and the capacity of the water-spaces requires to be considerably greater also, in order to allow for cleaning, and to facilitate the escape of the vapour through the dense fluid; so that perhaps one-fifth may be added altogether for the increased weight of water in the boilers, including the weight of the portion blown off. We must add, also, the rapid wear of boilers using salt water, and it will be apparent that an efficient method of surface condensation is one of the greatest desiderata of marine engineering.

The Foot Valve.—The valve which occupies the passage between the condenser and air-pump is called the foot valve, and is usually formed of brass in the manner shewn in Plates LX., LXI., where a leaf or flap valve is represented in section. This valve should have an area equal to one-third of that of the air-pump, and should be allowed to open to an angle of about 30°.

The Air-Pump.—The capacity of the air-pump is usually proportioned to that of the cylinder as 1 : 8. Other engineers adopt the rule of giving it the diameter of the cylinder, multiplied by ·56, with half its stroke, or a capacity proportional to this. As the reader has been already made acquainted with the work which the air-pump has to perform, we need not recapitulate it, but have only to observe that the smaller it can be made to do its work efficiently, so much the better. For sea-going vessels, the barrel of the pump is formed either of solid brass or gun metal, or of cast-iron lined with brass, the hot sea water rapidly corroding iron exposed to it. We prefer making it of solid brass, as the difference of expansion by heat, between the two metals, may cause the lining to *buckle* upon the cast iron, if not very well secured, and we have seen the brass bucket of the air-pump immovably jammed by its expansion exceeding that of the iron barrel containing it. The bucket and valves are formed of brass, the latter being usually what are called "butterfly valves," as represented in plan at Plate LXXVI., and in section at Plate LXXVII*. In the engine represented at Plates XIX., XX., as also in Plates LX., LXI., is shewn another description of bucket valve, formed by a brass disc encircling the air-pump rod, upon which it slides to admit the passage of the water round it during the descent of the bucket. This is technically called the "pot-lid valve."

Alloys of Brass.—The alloy here called brass or gun metal is usually formed by mixing together 2 lbs. of tin, 2 lbs. of zinc, and 16 lbs. of copper. Brass, for heavy bearings, 2½ lbs. of tin, ½ lb. zinc, and 16 lbs. copper. Brass for turning, 2 lbs. tin, 1½ lb. zinc, and 16 lbs. copper. Muntz's metal is 40 parts zinc and 60 copper; and Babbet's patent bushes for wheel axles or machinery are composed of 50 lbs. tin, 5 lbs. regulus of antimony, and 1 lb. copper. Bell metal is 4½ lbs. tin to 16 lbs. copper. Speculum metal about 8 lbs. tin to 16 lbs. copper. Brazing solders are formed of 3 lbs. copper, 1 lb. zinc, which is very hard; 8 lbs. brass, and 1 lb. zinc, which is not so hard; 6 lbs. brass, 1 lb. tin, 1 lb. zinc, which is a soft solder. A common proportion for soldering brass, iron, or copper, is equal parts of copper and zinc.

The Air-Pump Rod.—The air-pump rod is formed either entirely of gun metal or of iron cased in it. The latter is preferable, as it is stronger, but care must be taken that every part of the rod exposed to the salt water is covered with the brass, else the galvanic action which ensues will very quickly destroy the part left unprotected. As it is difficult to case the cone of the rod properly by the usual mode of plating with sheet brass, the plan of casting the brass casing entirely round the rod and cone has been adopted with success. The air-pump rod is usually made one-tenth of the diameter of the air-pump for iron, but if coated with brass, then one-eighth; if entirely of gun metal it may be made one-ninth, or of copper one-eighth.

The Delivery Valve.—The discharge or delivery valve, through which the contents of the air-pump flow into the hot well, is usually made of brass, and, like the foot valve, should have an area of one-third of that of the air-pump.

The Hot Well.—The cistern into which the hot water and air are delivered is called the hot well, from which the water is discharged into the sea at a little above the load water line of the vessel. The hot well is continued, however, for some distance above this, by means of a cast iron dome, or otherwise, from the top of which a small copper pipe is usually led through the ship's side for the escape of the air, which rises through the water when it reaches above the orifice of the discharge pipe. The principal use of the hot well dome is to provide a head of water, and an elastic air-spring, as it were, by means of which a continuous flow of water may be maintained through the discharge pipe during the descending as well as the ascending stroke of the air-pump bucket, which, it must be remembered, is single-acting. Hence, the area of the passage from the pump to the hot well should be equal to one-third of the area of the pump, but the orifice through the ship's side need not be more than one-sixth of the pump's area. In large engines, a sluice valve is usually fitted inside the vessel, across the mouth of the discharge pipe at the ship's side, which may be closed by hand when the engines are not working; thus preventing the wash of the sea from entering the hot well, which is

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consumption is 30 cwt. per hour, of which 1½ cwt. may be said to be due to the extra weight carried, and 1 cwt. for the work of the force-pumps—in all 2½ cwt.

further protected by a light flap valve hung over the mouth of the pipe outside.

The Feed Pumps.—A portion of the hot water, small in comparison to that removed by the air pump, is drawn by the feed pump from the condenser for the supply of the boiler, to restore the water evaporated into steam; as well as the amount which it is necessary to blow off into the sea, for the purpose of keeping the boiler clean by preventing undue saturation with salt or other mineral substances. Hence, the feed pumps must be proportioned to supply the loss from both sources. It is the practice in London to make them sufficiently large to deliver, each, from 80 to 100 cubic inches of water per horse power per minute, each pump being supposed large enough to supply the whole of the boilers, in case of accident to one of the pumps or feed pipes. Thus, ninety times the horse power of one engine, divided by the number of inches through which the plunger moves, in the descending stroke, during one minute, will be the area of the plunger in square inches. The proper quantity of feed is regulated by means of the feed cock on the front of the boiler, and the surplus water which is rejected by the feed is expelled into the sea (or the hot well) through an escape valve attached to any convenient part of the feed pipe, such valve being loaded with a small weight or spring. The feed pipes are formed of copper, and may have one-third of the area of the plunger. The feed pumps are of cast iron, and are usually fitted with a brass plunger, which is simply a rod, or ram, cast hollow for lightness, and sliding through a stuffing box fitted with brass bushes, as represented in Plates LXIII. and LXXVII*. In the engines represented by the plate last mentioned the plunger receives its motion from the air pump cross-head in such a manner that it may be connected or disconnected at pleasure during the working of the engines, which is a good plan where one feed pump is sufficient to supply all the boilers. The plunger rod may be made of iron, one-third of the diameter of the plunger.

The Bilge Pumps.—Bilge pumps are invariably attached to marine engines, both as a security to the ship in case of extraordinary leakage, and a constant saving of labour to the crew. These pumps are made of a corresponding size and construction with the feed pumps, one bilge and one feed being attached to each engine. The bilge pipes should be made of lead, which suffers less than copper from the action of the acid contained in the bilge-water of wooden ships; and care must be taken that they do not get choked with filth.

Brine Pumps.—Until recently, brine pumps were very generally attached to engines using tubular boilers, for the purpose of extracting the brine in a more regular manner than by blowing off, and for making this important operation independent of the engineer, as well as for economizing a portion of the heat thus thrown away by transferring it to the feed-water. This is accomplished by passing the feed through a series of tubes enclosed in a small cistern, (miscalled a refrigerator), through which the brine is forced by the pumps before entering the sea. The use of this apparatus, however, is hardly to be recommended, from the liability of the tubes to get choked with salt or deposit; the more simple method of blowing off through a pipe and cock communicating directly with the sea being now usually preferred as the safer plan.

Hand-Pump in Engine Room.—A pump must also be fitted in the engine room, for the purpose of feeding the boilers when the engines are at rest, and the steam is blowing off. In the case of vessels with tubular boilers, this pump usually receives its motion from a little high pressure engine (technically called a *donkey*), which works by the pressure of steam in the large boilers. This is required on account of the rapid evaporation of tubular boilers in comparison to their area at the water level; but in the case of flue boilers, the water surface being comparatively larger, and the risk of the water level falling too low during the temporary stoppage being proportionably diminished, this pump is generally worked by hand, when required for feeding or for filling the boilers above the sea level, before the fires are lighted.

The feeding of flue boilers by hand may be obviated, in most cases, by allowing a little more feed than usual to enter the boiler, preparatory to a stoppage. The hand-pump, however, should be always made capable of being connected, and that instantly, with the engines; so as either to assist in feeding if required, to act as a fire engine, in case of fire, or for the every-day duty of washing decks. This pump should be made capable of drawing either from the sea, the bilge, or the boiler.

Cocks and Valves.—A good proportion for the taper of conical brass cocks is two inches to the foot, and they should not be made too light in metal, (if large,) otherwise the pressure may cause them to leak. Conical spindle valve faces are turned to an angle of 45°.

The Cross-heads.—The piston or air pump cross-head should have a breadth across the centre boss of twice the diameter of the rod, and a depth of $2\frac{1}{2}$ times. If for a side lever engine, the thickness of the blade may be made to taper from $\frac{1}{13\cdot5}$ of the diameter of the cylinder next the boss, to $\frac{1}{15}$ at the necks. The necks or bearings should be each of the same diameter with the rod.

Side-rods.— The piston or air pump side rods, at the thinnest part, should be each three-fourths of the diameter of the rod, and should swell in the middle one-fourth inch to the foot. The thickness of iron at the side of the eye to be the diameter of the bearing in inches, divided by 2·1, and at the crown by 1·85.

The Cross-tail.—The cross tail should be proportioned from the cross head, each dimension being made stronger by one-eighth part. The cutters for cross-heads and cross-tails to taper one-fourth of an inch to the foot.

Straps, Gibs and Cutters.—For the general thickness of straps at the crown and at the cutter hole, divide the diameter of the bearing by 2·5, and for their thickness at the thinnest part by 3·2. Gib and cutter holes should have a length equal to the diameter of the bearing, and a breadth of one-fourth of the length; or their area should be made equal to one-fourth of the sectional area of the bearing. The distance from the end of the cutter hole to the end of the strap should be $\frac{1}{1\cdot65}$ of the diameter of the bearing, and from the other end of the cutter hole to the butt end of the rod, half the diameter of the bearing. The breadth of the strap should be one-eighth part less than the diameter of the bearing; the width within the strap being equal to the bearing, plus the double thickness of the brass, which usually varies at the sides from one-half to five-eighths of an inch thick for the larger sizes of bearings. Gibs and cutters for connecting rods, side rods, &c., should taper three-fourths of an inch to the foot; and keys three-sixteenths of an inch to the foot. For the depth of nuts, multiply the diameter of the screw by 1·15; and for the breadth across the flat, multiply by 1·8. A table of the pitches of screws, employed by a celebrated engineer, will be found at page 175.

The Connecting Rod.—The connecting rod should have a diameter at the thinnest part of $\frac{1}{8\cdot5}$th, or one-ninth the diameter of the cylinder; and may be made to swell in the middle one-fourth of an inch to the foot of length.

The Side Levers.—The main centres for the side-levers should have the diameter of the cylinder in inches divided by 6·75. The depth of a side lever, (of cast iron), in the centre may be found as follows:—Multiply the length of the lever in feet by ·75, and extract the cube root of the product. Then square the diameter of the cylinder in inches, and extract the cube root also. Multiply together the two roots thus found, and the final product will be the depth of the side lever in inches at the centre.

For the diameter of the side lever outer studs, multiply the diameter of the cylinder in inches by ·07.

The Crank Shaft.—For the diameter of crank shaft in inches, a good rule is to multiply the nominal horse power by 320, and extract the cube root of the product; and then divide the root thus found by the number of strokes per minute. Another common rule is to allow one square inch of area per horse power. The shaft begins to taper after passing the first set of paddle wheel arms to the outside bearing, which may be made two-thirds of the in-board diameter.

The Intermediate Shaft.—The intermediate shaft for sea-going vessels should have a diameter in the centre, found by multiplying the diameter of the cylinder by ·22; and for river navigation by ·2: the bearings being the same as for the crank shaft.

The Crank Pin.—The area of the crank pin should be one-half the area of the crank shaft: some engineers make its diameter one-seventh of that of the cylinder.

The Crank.—The diameter across the large boss of the crank may be found by multiplying the diameter of the shaft by 1·8; and the length of the boss may be made one-tenth less than the diameter of the shaft. The small boss of the crank to be proportioned in the same way from the pin. The breadth of the web will be found, for the broad end, by multiplying the diameter of the shaft by 1·16, and by making the breadth at the small end equal to nine-tenths of the diameter of the shaft. The thickness of the web is found by multiplying the diameter of the shaft by ·55.

The Parallel Motion.—The system of parallel motion rods usually applied to a marine engine is represented at Fig. 6, in Plate XIII., in the description of which will be shown the method of finding the centres and lengths for the radius and parallel bars, &c. We would add here, however, the following simple and practical rules for determining the lengths of the parallel motion rods, given by Mr. Hann of King's College:—

Add the distance between the vertical line and the axis of the radius rod to twice the radius of the beam. Divide the square of the radius of the beam by the above sum, and the quotient will be the length of the parallel bar.

If the radius rod be shorter than the parallel bar, divide the square of the radius of the beam by the difference, instead of the sum.

The distance between the vertical line and the axis of the radius rod, added to or subtracted from the parallel bar, according as the radius rod is longer or shorter than the parallel bar, gives the length of the radius rod.

Or, add the radius of the beam to the distance between the vertical line and axis of the radius rod; divide the square of this sum by twice the radius of the beam added to the above-named distance, for the length of the radius rod.

When the length of stroke is taken into consideration, subtract the square of half the length of the stroke from the square of the radius of the beam, and extract the square root of the remainder; to this root add the above radius of the beam, and multiply this sum by half the radius of the beam, for a dividend. Then add the above root, the radius of the beam, and the distance between the vertical line and the end of the radius rod together for a divisor; and if the above dividend be divided by this divisor, the quotient will be the length of the parallel bar; and this added to the distance between the vertical line and the end of the radius rod will give the length of the radius rod.

Holding-down Bolts and Foundation.—The holding-down bolts are usually made of wrought-iron, but occasionally of copper or Muntz's metal, to prevent corrosion, to which iron bolts are very liable. They usually pass both through the keelsons and the ship's bottom, and are secured by a head or a nut underneath, and a nut on the base plate. The plan of screwing them into the timber keelsons, as wood screws, has been adopted with success, when used in this way along with some through bolts on the old principle. The strain to which a bolt may be safely exposed is about 4000 lbs. to the square inch of area, (without reckoning the portion cut by the thread), but, of course, large allowance must be made for corrosion, inequality of strain, &c., especially in the case of the holding-down bolts. In iron vessels these bolts are secured by a nut underneath the top plate of the hollow iron beams which support the engine. The base plate of the engine should be soundly bedded upon a solid platform of hard timber, (teak is the best,) or at least upon broad and substantial engine bearers, and the bottom of the cylinder, if it forms a part of the base plate, must be protected from the wash of the bilge water which would otherwise condense the steam inside.

Engine Framing.—The entablature of the engine which supports the paddle shaft is still usually formed of cast-iron, although wrought iron has been recently introduced here with much advantage, in order to save weight and keep the centre of gravity as low as possible. A wrought-iron entablature is also less liable to be cracked by the working of the side of the ship when she is labouring in a heavy sea, which is by no means an uncommon accident in the Royal Navy. Gun-metal is also coming into use for this purpose, and makes a very light and elegant entablature. The pillars supporting the entablature are now almost universally formed of wrought-iron, extreme lightness being the distinguishing feature of the most recently constructed engines. The old description of side-lever engine with flue boilers and paddle wheels, weigh not less in general than one ton per horse power with water in the boilers, and sometimes more; but by the innovation of direct-acting engines and tubular boilers we are now enabled to construct the whole of the machinery with water in boilers, at the rate of 12 cwt., and in the case of oscillating engines even 10 cwt. per horse, thus saving half the weight of the engines under the old regime, for coals or merchandise; and by diminishing at the same time the length of the engine room, providing a greater accommodation for passengers.

The Paddle Wheel: Varieties.—It is now the general opinion of practical men that the common paddle wheel, if well proportioned and not too deeply immersed, is preferable, under ordinary circumstances, to any of the "improved" varieties which have from time to time appeared. The only one of these which has at all stood the test of experience is Morgan's feathering paddle-wheel, by using which the breadth and height of the paddle boxes may be somewhat diminished; but it is objectionable from its weight, which is nearly twice that of the common wheel, from its additional expense, and from its complexity and consequent liability to derangement. Mr. Fields' cycloidal wheel, in which the paddle board is divided into steps with an open space between each, can hardly claim any superiority over the undivided board, but the simple modification of dividing the board into two breadths, placed one on either side of the paddle arm, is extensively adopted in the larger class of steam vessels, and appears to answer well.

Dip of the Wheel.—The grand objection to the paddle-wheel is the unequal "dip," or immersion of the boards, consequent on the varying draught of water of the steam ship, as her coals and stores continue to be consumed on the voyage. Thus a vessel with a displacement of 12 tons per inch at the load line, consuming 460 tons of coal, will swim 3 feet 6 inches lighter at the end of the voyage than she did at the commencement, so that it can only be for a short time during the middle of the voyage that her wheels can have the dip best calculated for their due performance. The most advantageous dip varies with the diameter of the wheel, the velocity with which the vessel is propelled, and the service in which it is employed as for sea or river navigation. For sea-going vessels of the first class this may be taken at 18 to 21 inches over the boards when vertical, and for the smaller class of steamers, at 12 or 15 inches. It is the practice in the fast river boats on the Thames to allow only a very few inches over the vertical board when the wheel is large. The *slip* of the paddle wheel, or in other words, the excess of its velocity above that of the vessel, is usually reckoned at one-fifth (or 20 per cent.) of the vessel's speed, when the wheel is properly proportioned and the vessel has tolerably good water lines for speed.

Area of the Paddle Boards.—In proportioning the area of the paddle boards, we must, of course, take into consideration the estimated dip of the wheel. By taking an average of the practice with regard to large sea-going steamers, we find that the whole paddle-board surface for both wheels varies from $2\frac{1}{4}$ to 3 square feet per horse power; and for river steamers, with light dip, about 5 square feet. The surface of each board, for large vessels, rarely exceeds $\frac{1}{25}$ of the total horse power in square feet, but increases, as the dip and diameter of the wheel diminishes, to one-fifteenth; while, for river steamers, one-tenth of the total horse power in square feet is not an unusual proportion. A long narrow board is of course more effective than a short and deep one, but is inconvenient, especially in sea-going vessels, from the increased breadth of the paddle box. Hence, in large steamers, the usual proportion of depth to length varies from one-fourth to one-fifth; but for river steamers the average length is about seven times the depth of the board.

Disconnecting Apparatus.—As it is sometimes desirable, (especially in the Government service), to disconnect the paddle wheels from the engines, and thus permit them to revolve in their journals while the ship is sailing under canvas, several expedients with this view have been adopted, although none, hitherto, with much success. That generally employed by the Admiralty is Braithwaite's apparatus, which is represented in the annexed figures and may be thus described:—A cast iron concentric disc *a* is keyed on the paddle-shaft *b*, and a strong wrought iron hoop *c*, lined with brass, surrounds and slides on the periphery of the disc. On this hoop a projection *d* is forged, in which one end of the crank pin is securely keyed, the hoop thus taking the place of the outer crank, and at another place it is swelled out to receive a brass cushion *e*, placed between the hoop and the disc before-mentioned. The cushion is now driven tight against the disc by means of a key through the swollen part of the hoop, and by its friction connects the hoop *c* with the disc *a* fixed on the paddle-shaft, which now revolves, and communicates motion to the wheels. The latter are instantly disconnected by driving back the key and thus loosening the friction cushion, when the hoop and the disc revolve independently of each other. This disconnecting apparatus is by no means a perfect one, and gives much trouble at sea by the slipping of the hoop, although the key may have been driven as tight as is practicable; and the hoop itself has frequently given way, the forging being rather a difficult one to make perfectly sound. Messrs. Maudslay adopt the plan of sliding the paddle shaft

Fig. 8.

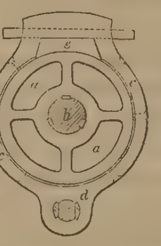
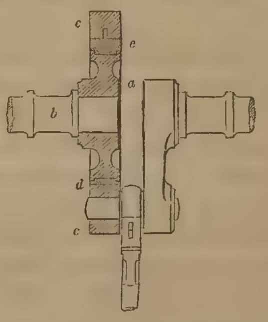


Fig. 9.



bodily, with the outer crank upon it, out of the crank pin, by means of a worm wheel and cogged collar on the shaft, which is worked from the deck. Messrs. Seaward, again, form the pin end of their outer crank with a slot cut out of the thickness of the boss on the inside, the breadth of the slot being equal to the diameter of the pin, which is held fast by square keys while the engines are working. When it is wished to disconnect the wheels, the engines are stopped, and the keys withdrawn, when the boss of the outer crank, at each revolution of the wheel by resistance from the water, clears and passes the pin by means of the slot.

Proportions of Paddle Wheels.—As any attempt to reduce the properties of the paddle wheel to the precision of mathematical calculation has always proved as futile in practice as it is complicated in theory, we prefer subjoining the following table of the proportions of paddle wheels, as reduced from the practice of our best constructing engineers:—

NAMES OF ENGINEERS.	Total Horse Power.	Length of Stroke of the Engines.	No. of Revolutions per minute.	Diam. of Wheels to outside of Boards.	Breadth of Wheels.	Estimated Dip to under side of Boards.	Ratio of diam. of Wheel to Crank.	Velocity of outsides of Boards in feet per minute.
	Horses.	Ft. In.	No.	Ft. In.	Ft. In.	Ft. In.	Ratio to unity.	Feet.
Boulton and Watt, ...	480	5 6	20	26 0	9 0	4 2½	9·485	1641·5
Bury, Curtis & Co.,....	425	5 6	20	25 6	9 0	4 0	9·273	1522
Butterley Co., ...	420	5 4	20	26 0	8 6	4 0	9·593	1602
Caird & Co.,.............	500	5 6	—	25 6	9 6	—	9·563	—
Fairbairn & Co .,.......	460	5 3	20	26 0	8 6	4 2½	9·984	1646·7
Forrester & Co., ...	460	7 6	16	29 6	9 6	6 0	7 889	1487
Fawcett & Preston,....	445	5 9	19	26 0	9 0	4 9	8·125	1446·5
Maudslay, Sons, & Field,	450	6 0	18	26 9	10 0	4 5	10·763	1538
Miller, Ravenhill, & Co.,	545	6 6	17½	25 0	10 6	3 9	7·718	1379
Napier, Robert, ...	500	5 6	20	26 6	9 4	—	9·938	1665
Penn and Son,...........	500	6 0	18½	27 0	9 0	—	9·014	1571·6
Rennie, G. & J.,........	500	5 8	19	20 6	9 0	4 4	9·353	1602·6
Rigby, John, ...	510	5 9	19¼	24 6	9 0	3 6	9·046	1462·4
Scott, Sinclair & Co.,...	342	5 0	18	25 0	8 0	3 8	10·033	1418·4
Seaward and Capel,....	470	5 4½	21	26 6	9 6	4 6	9·860	1748·3

Specification for a Pair of Paddle Wheel Engines.—We subjoin the specification for a pair of 60 horse paddle wheel engines as constructed for H. M. Service.

The engines to be constructed on the oscillating principle, embodying all the latest improvements.

The collective power to equal 120 horses.

Weight of the engines, paddle wheels, and spare gear not to exceed..........................42 tons.
— boilers and coal boxes....................21 —
— water in boilers............................15 —

Total weight of machinery.........................78 tons.

Diameter of cylinders to be...................................43 inches.
Length of stroke.. 4 feet.
Strokes per minute..31.
Diameter of paddle wheels.................................19 feet.
Breadth of do. .... 7 ft. 6 in.
Diameter of necks of intermediate and paddle shafts, 9½ inches.

Coals stowed in coal-boxes to be 50 tons, or 5 days' consumption at 8 lbs. per horse per hour.

The cylinder valves will be of solid gun-metal, of the most approved construction, with springs.

There will be one air pump of solid gun-metal, worked from a crank forged on the intermediate shaft, and the air pump rod and bucket, foot valve, and delivery valve, will be all of gun-metal.

There will be two feed and two bilge pumps, each capable of supplying both the boilers.

The pistons will be of the most approved construction, with metallic packings and springs.

Expansive gear will be attached to the engines.

All steam pipes, feed pipes, and others usually formed of copper, will be of this material of sufficient thickness, and all bearings will be of best brass.

The framing of the engines will be composed entirely of wrought-iron, and will be made as light as is consistent with the requisite strength.

The space occupied by the engines in the length of the engine room will be 7 feet, 6 inches.

The paddle wheels to be constructed on the ordinary principle, the shaft running through to an outside bearing.

The capacity of the coal boxes to be estimated at 48 cubic feet to the ton, a space of 6 inches under the deck being excluded in the computation.

The holding-down bolts will be secured by nuts let into the lower sleepers, so as not to require the bolts to pass through the vessel's bottom, and they will have at their lower ends one-inch wrought-iron washers about 8 inches square, interposed between the nuts and the wood.

The steam pipes to be ¼ inch thick, the bilge pipes ¼ inch, the feed pipes ⅛ inch, the waste steam pipe of copper ⅛ inch, and the discharge or waste water pipes of copper ¼ inch thick.

The cylinder will be fitted with discharge or escape valves at top and bottom, the valves being fitted with suitable metallic cases or hoods.

The threads of all screwed bolts, nuts and pins, used in the engines and boilers, will agree with the standard screws employed by the steam Navy.

A small engine will be fitted for working one of the pumps for feeding the boilers.

Pipes to be fitted for supplying, in the event of a leak in the vessel, the requisite quantity of water from the bilge to the condensers.

The hand pump will be made capable of being worked by the engine also, and will be arranged to pump into the boilers, on deck, or overboard; and will be capable of drawing water from the boilers, the bilge, or the sea.

All the necessary ladders for engine room, with fenders, guard rails and floor plates to be included in the tender, and likewise the expense of trying and fitting the spare gear.

The cylinders will be clothed with hair-felt to the thickness of 2 inches; this will be covered with thoroughly dried wood, well fitted and bound together by iron or brass hoops.

The steam pipes will be clothed with felt, bound round with spun yarn, and covered with canvas; the whole being of such thickness as to be even with the outsides of the flanges.

The boilers will be two in number, 14 feet wide and 9 feet 6 inches long, on the tubular principle, and embodying all the latest improvements.

The tubes will be of iron (patent welded) 3 inches outside diameter and 6 feet long, being secured in the tube plates in the best manner, without ferrules.

The tube plates will be of the best charcoal iron ⅝ inch thick.

Bottoms of boilers will be composed of plates 7/16 inch, and the sides and tops of plates ⅜ inch thick, of the best Staffordshire iron.

Fire boxes will be formed of Lowmoor iron ⅜ inch thick.

The heating surfaces of the boilers will be as follows, viz., grate-bar surface 85 square inches per horse power, and tube and flue surface together to be 13½ square feet per horse.

Sufficient room will be left between the crown of the furnace and the tubes, for cleaning or scaling the former.

Each boiler will be fitted with separate safety valve and stop valve, so that either may be worked independently of the other, if required.

Brine pumps and refrigerators will be attached to the boilers, in addition to the usual blow-off cocks.

The blow-off pipes will be 3½ inches diameter and ¼ inch thick, formed of copper, and having their cocks, conical pipes and stop valves of gun-metal.

A separate damper to be fitted in the bottom of the funnel for each boiler, and dampers to be fitted to the fronts of the ash pits.

All mud-hole doors will be fitted *inside* the boilers.

A space of 13 inches to be left clear between the boilers and the coal boxes in every part.

The boilers to be bedded on mastic, as usual in the Government service.

The boilers, after having been proved and painted, are to be clothed with felt, previously stitched to canvas. The felt to be two inches thick and applied to the tops, sides and backs while the paint is still moist. The canvas outside is then to be painted and covered with thoroughly dried 1 inch deal boards, with rabbeted or groove-and-tongue joints, and bound to the boilers by suitable iron straps. The felt and boards to be kept at least 18 inches from the funnel, and the circular space left is to be covered with a 3 inch course of brick, set in cement, and surrounded by an iron hoop, or the space may be filled with mastic. The boards and bricks on the top of the boilers to be covered with sheet-lead, 4 lbs. to the square foot, so as to prevent any leaks from the deck reaching the felt.

The horse power to be calculated at 7 lbs. effective pressure per square inch of piston, and the speed of the piston to be assumed at 196 feet per minute.

All expenses for coals, &c., until the engines are complete, will be borne by the contractors.

The following list of spare gear and duplicates will be supplied with the engines, viz.—

Engineers' tools, comprising—
  25 brushes for boiler tubes.
  2 drifts, short and long.
  12 fire irons.
  2 mandrils.
  6 scrapers, circular and forked.
  24 spanners and wrenches of sorts.
  1 set of stocks, taps, and dies, from ¼ to 1½ inch.
1 air pump rod and cross head.
4 arms for paddle wheels.
½ set of furnace bars.
3 bearers for bars.
1 bilge pump rod.
6 cwt. boiler plates.
120 bolts and nuts for engines, properly assorted.
100 bolts and nuts (or rivets) for paddle wheels.
100 hook bolts and nuts for paddle boards.
2 brasses for outer end of shaft.
1 piston rod and brasses complete.
1 cylinder lid with gland complete.
1 feed pump.
1 foot valve without seat.
12 paddle boards, and 18 paddle plates.
1 piston complete.
1 slide rod.
4 large segments for paddle wheels and 2 small do.
1 set of springs for each piston.
1 set of springs for other parts, complete for one engine.
12 boiler tubes.
2 glass tubes for barometers.
4 glass tubes for boiler water gauges.
100 iron washers.

Time required for completing the work ready to be put on board, ... Six months.
Further time required for fixing complete, . . . Six weeks.

Proportions of Marine Engines.—In the Appendix will be found a table of proportions, by different makers, for a pair of paddle wheel engines corresponding to an estimated weight of 300 tons. Here we have also introduced practical tables of dimensions of marine engines of different sizes, carefully measured from existing samples by Messrs. Fairbairn and other eminent engineers. At page 25 we have given comparative views of some of the best descriptions of direct acting engines which have hitherto been proposed, in addition to those illustrated by Plates LXXIV. to LXXVII*.

Before quitting the subject of marine engines, which our limited space compels us to treat only in a cursory manner, we would offer a few remarks on the action of the screw propeller, and its applicability to the purposes of steam navigation.

Comparison of the Modes of Propulsion by Paddle Wheels and Screws.—On a *primâ facie* view of the matter, the advantages which the screw would appear to possess over the paddle wheels as a means of propulsion are overwhelming, but if we call to our aid the experience and opinion of practical men, we shall find that the screw has, of late years, progressed but little in public estimation, if it has not actually retrograded. Let us examine, therefore, into the merits of this much agitated case of screw *versus* paddle wheel, and consider impartially the various arguments that have been adduced on both sides of the question.

Apparent Advantages of the Screw in a Merchant Steamer.—In the case of a merchant steamer, the comparative advantages of the screw may be stated thus:—1st, the absence of the paddle boxes, by which the resistance of the wind to the vessel's hull is diminished, and the sails are more readily managed; 2ndly, the low position of the machinery in the hold, by which means nearly the whole of the main deck may be made available for passenger accommodation; 3dly, an increased stability in the vessel, resulting from the low position of the centre of gravity of the machinery, so that the top sides may be either raised higher than would otherwise be safe, or a heavier rig, or armament, may be adapted to the vessel; 4thly, a more uniform action of the propelling power during a long voyage, the efficiency of the screw being but little impaired by the variation in the draught of water, which is so prejudicial to the proper performance of the paddle wheel; 5thly, the more uniform action of the propelling power when the ship is *rolling*, in which case the paddle wheel is at one moment deeply immersed and then left dry alternately, causing a serious loss of speed to the vessel, and a dangerous strain upon the machinery; 6thly, a diminution in the weight and expense of the vessel's hull, owing to the absence of the cumbrous paddle beams and boxes, &c.; 7thly, a probably increased length of duration in the structure of the vessel, from the vibration being less, and confined to the lower and stronger portion of the hull.

Apparent Advantages of the Screw in a Vessel of War.—In the case of vessels of war, we have all the advantages already enumerated for the screw, with the following in addition:—8thly, an increased broad-side for guns both on the upper and main decks; 9thly, the comparative security of the machinery and the propelling agent from the enemy's shot, owing to their position under the water line; 10thly, the facility with which the screw can be disengaged from its bearings, and hoisted up out of the water, when the ship is under canvas; and, lastly, the facility which the screw affords for disguising the true character of the vessel, when the sails are set, and the telescope funnel dropped under hatches. Thus ends the case for the prosecution, and a formidable one it certainly is.

Actual Results hitherto obtained from the Screw.—On the other side, the paddle wheel, as defendant, maintains that, notwithstanding all these advantages, its rival has not succeeded in establishing any superiority in practice, but has hitherto shown an inferiority in that all-important point of *speed*, to which every other qualification in a merchant steamer must be subservient. We allude more especially to those cases in which the screw has lately been applied in the Government service, and although fully admitting that it would be unjust to tax it individually, at this early stage of our acquaintance, with failures which have evidently resulted in a great measure from our want of experience in its *application*, still we doubt whether the results that have hitherto been obtained from it augur well for its general performance at high rates of speed, when applied to wooden vessels. In the cases to which we have just referred, it is now ascertained that the principal cause of failure has been owing to an undue fulness of the after lines, or "run," of the body of the vessel, which causes the screw aperture to be bounded on the fore side by a flat surface, or "square tuck," impeding the free access and escape of the water, and entailing a heavy drag of dead water upon the ship. It has been proved by direct experiment with the "Dwarf," Admiralty steam tender, that any filling out of the lines immediately before the screw has a most injurious effect upon the speed of the vessel, and several of the Admiralty screw steamers, which have lately had their lines fined away at that place, have advanced from a speed of about six knots, to a speed of ten, or perhaps eleven knots an hour. This is still considerably short of what experience would lead us to expect from the same power of engines applied to paddle wheels, but large allowance should be made for inexperience in this new mode of propulsion.

Uncertainty in the Performance of the Screw.—Owing, doubtless, to our present imperfect knowledge of the screw, it appears uncertain and capricious in its action; or, in more correct language, it is found that the particular form of the vessel's run, the shape and mode of finishing of the aperture, and some other practical points connected with its application, exert a remarkable influence, (which is not yet thoroughly understood), upon the performance of the screw, and the consequent speed of the vessel. We only know for certain that the after lines of the vessel should be kept as fine as possible just before the screw, and that the body should virtually terminate at the inner stern post, so as to avoid the flat surface or square tuck already mentioned. In wooden vessels, the boundary of the screw aperture is usually sheathed and strengthened by a gun-metal casting, by means of which a sharper finish can be given to it. In this point of view iron vessels have shown a decided superiority over wooden ones for the adaptation of the screw propeller, owing to the comparative thinness to which they may be tapered at the stern post with a due regard to strength. As considerations of this nature do not detract from the actual power of the screw, they will cease to be objections so soon as we learn from experience that they can be overcome by a better mode of application; but until this point has been satisfactorily arrived at, it cannot be admitted with justice that the screw is equally effective with the paddle wheel, although it appears neither impossible nor improbable that it may soon become so. For vessels of war, there can be no doubt but that the screw will prove a most valuable auxiliary, even although the speed obtained may not be quite equal to that from paddle wheels. For river navigation, where the draught of water is limited, the screw does not appear very applicable, as a large submerged diameter has been proved to be an essential requisite for its due performance. As an auxiliary power to sailing vessels making long voyages it seems, on the other hand, to be peculiarly adapted, and we believe the day is not far distant when its application to this class of merchantmen will be the rule in place of the exception.

Although the action of the screw is not much impaired by the *rolling* motion which proves so detrimental to the power of the paddle wheels, it suffers considerably from the *pitching* motion of the vessel, in seas where the waves run in a direction perpendicular to the vessel's course. This is owing to the position of the screw-propeller at one extremity of the axis of rotation, so that a large portion of its diameter is alternately left dry and deeply submerged as the vessel pitches; whereas the paddle wheels, being situated near the centre of rotation, are not seriously affected by this motion. It will be observed that the pitching motion is most felt when the vessel is going head to wind, which is just the time when it is so desirable to obtain their full power from the engines.

Gearing applied to the Screw.—As it is essential for the due performance of the screw that it should revolve with a considerable velocity, it has been deemed necessary to introduce gearing to multiply the speed of the engines, in those cases where a high speed is looked for from the vessel. The use of gearing is highly objectionable in sea-going vessels, as it is liable to be broken or deranged by the sudden shocks received by the screw in a rough sea. With the view of affording additional strength and security against breakage, Mr. Fairbairn has introduced the method of internal gearing represented by the annexed cuts.

Fig. 10.

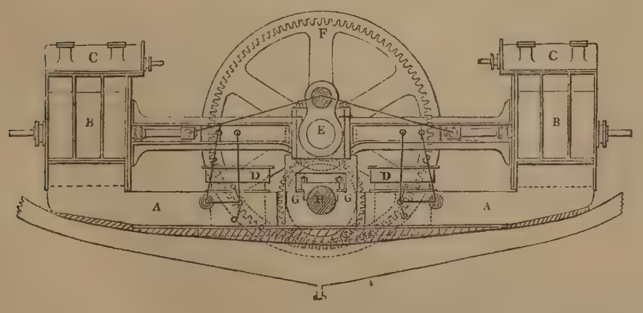
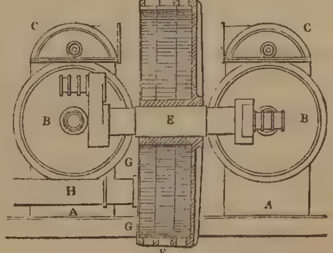


Fig. 11.



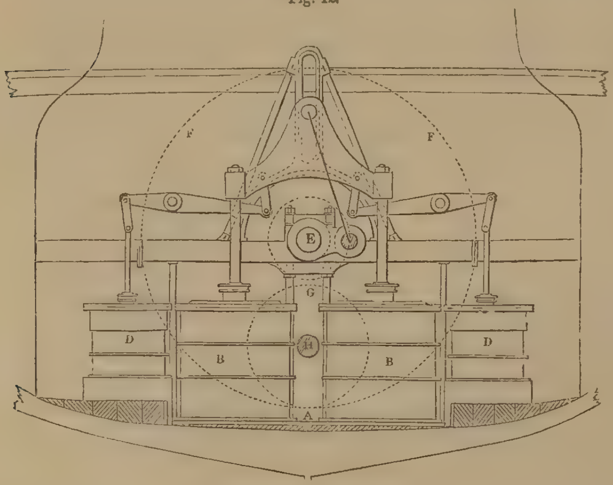
Screw Engines, with Internal Gear.—Scale, $\frac{1}{8}$ inch to a foot.

Fig. 10 is a transverse, and Fig. 11 a longitudinal elevation of the arrangement here referred to, as adapted to a set of four horizontal cylinder engines, of the collective power of 556 horses, constructed for the Government service. A, A, denote the sole plates and condensers. B, B, the steam cylinders, having their piston rods protruding at both ends. C, C, the valve casings. D, D, the air pumps. E, the crank shaft, carrying the great internal cogged spur wheel F, which gears with the pinion G, on the propeller shaft H. The strong lines in Fig. 10 shew the arrangement of rods for the transmission of motion.

The advantage of this plan lies chiefly in the greater number of teeth which are thus brought into gear at one time, so that the strain is divided amongst several, in place of being wholly transmitted through one tooth. The arrangement is also compact, and admits of a high velocity being communicated to the screw shaft without raising the crown of the driving wheel above the water line of the vessel, or introducing such ruinously small wheels as some of those recently constructed for the Admiralty.

Fig. 12 represents another description of engine adapted either for internal gearing or for driving the screw direct. The same letters of reference employed in the preceding example are here used to indicate the corresponding parts.

Fig. 12.



Engine for Screw Propeller.

The centre of the screw shaft, as shown in the wood-cuts 10 and 12, is considerably below the axis of the screw at the stern, to which the shaft inclines gradually up from the engine room, in order to keep the machinery there as low as possible.

Practical Details of Screw Engines.—As the readiest means of giving some practical information on the details of screw machinery, we subjoin the specifications for several pairs of engines adapted for screw propulsion in H.M. service.

Specification I.

Collective power of the engines	700 horses.
Weight of the engines	175 tons.
— boilers and apparatus	80 —
— water in boilers	60 —
— coal boxes	19 —
— propeller shaft and gearing	56 —
— spare gear and fittings	30 —
Total weight of machinery with water	420 tons.

The engines to be constructed with horizontal cylinders, eight in number, placed in pairs, with one double slide valve to each pair, and one connecting rod, the two opposite connecting rods working on the same crank pin. By this arrangement the whole of the machinery is kept several feet below the water line. Toothed gearing is employed to give the requisite speed to the propeller shaft, the large wheel being fitted with wooden cogs (of hornbeam) and the pinion of cast-iron carefully pitched and trimmed. The air pumps to be worked by levers connected with piston cross heads, and to be made of gun metal as usual. Expansion gear to be fitted to the engines.

Diameter of cylinders, (8 in number) each	51 inches.
Length of stroke	3 ft. 6 in.
Strokes of engines per minute	29
Revolutions of propeller per minute	77
Diameter of necks of engine shaft	16½ inches.
— — propeller shaft	15½ inches

The screw to be of gun metal, double-threaded, common pitch, 14 feet diameter, 18 feet pitch and 3 feet 9 inches long.

The weight of coals carried in the length of the engine room, under the main deck, to be 280 tons. Estimated consumption of coal, per horse per hour, 7 lbs.

The boilers to be four in number, two abaft and two before the engines, and the tops of the steam chests to be about four feet below the water line of the ship.

Time required for completing the whole of the machinery, nine months; and three months additional for erecting.

Specification II.

Collective power of the engines	620 horses.
Weight of the engines	162 tons.
— boilers and apparatus	73 —
— water in boilers	55 —
— coal boxes	16 —
— propeller, shafts, &c	49 —
— spare gear and fittings	25 —
Total weight of machinery	380 tons.

The engines are designed to drive the screw direct, employing twelve horizontal cylinders working on three cranks on the propeller shaft, two connecting rods to each crank pin.

Diameter of the cylinders (12 in number) each	36 inches.
Length of stroke	2 feet.
Number of strokes of engines per minute	60
Revolutions of screw per minute	60
Diameter of necks of screw shaft	15 inches.

The screw to be gun metal, common pitch, double-threaded, 14 feet diameter, 19 feet pitch, 3 feet 3 inches long. To carry 420 tons of coals, or nine days' consumption at 7 lbs. per horse per hour.

Specification III.

Collective power of the engines	520 horses.
Weight of the engines	127 tons.
— boilers and apparatus	62 —
— water in boilers	48 —
— coal boxes	16 —
— propeller, shaft, and gearing	37 —
— spare gear and fittings	25 —
Total weight of machinery with water	315 tons.

The engines are designed to transmit the power to the screw shaft through internal gearing, and employ four horizontal cylinders (two on each side of the centre line) with a connecting rod to each cylinder, the two opposite rods working upon the same crank pin on the engine shaft. The air pumps are worked by levers from the piston cross heads. (See woodcuts, Figs. 10 and 11.)

Diameter of the cylinders (4 in number) each	59 inches.
Length of stroke	3 ft. 3 in.
Number of strokes of engines per minute	34½
Revolutions of screw per minute	85
Diameter of necks of engine shaft	14 inches.
— — screw shaft	12½ inches.

The screw to be double-threaded, 12 feet 6 inches diameter, 15 feet pitch, 2 feet 9 inches long.

Coals carried in engine room, under the main deck, 225 tons; on part of the main deck, between the engine room bulkheads, 250 tons. Total coals carried, 475 tons or twelve days' consumption.

Specification IV.

Collective power of the engines..............................400 horses.
Weight of the engines.................................. 97 tons.
— boilers and apparatus................. 45 —
— water in boilers........................ 36 —
— coal boxes.............................. 14 —
— propeller and apparatus............ 28 —
— spare gear and fittings............. 20 —

Total weight of machinery with water...240 tons.

The engines are designed with four horizontal cylinders on one side of the centre line of the vessel, balanced by the driving wheel, with outside gearing on the other.

Diameter of the cylinders (4 in number) each....51½ inches.
Length of stroke... 3 ft. 3 in.
Number of strokes of engines per minute...........34
Revolutions of screw per minute.....................85
Diameter of necks of engine shaft....................12½ inches.
— — screw shaft.....................11 —

Screw, 11 feet 6 inches diameter, and 2 feet 6 inches long. Time for completing machinery, nine months; for erecting, ten weeks.

Specification V.

Auxiliary power for a steam guard ship of 72 guns.
Collective power of the engines.......................450 horses.
Weight of the machinery and water.................280 tons.

The engines to have four horizontal cylinders, two on each side of the shaft, driving direct on two crank pins; the weight of the piston being, in each case, supported by a rod passing steam-tight through a gland in the cylinder bottom. (See Fig. 10.)

Diameter of the cylinders (4 in number) each......50 inches.
Length of stroke.. 3 feet.
Number of strokes of engines per minute..............45
— — screw propeller.................45
Diameter of necks of screw shaft.......................13 inches.

Coals in boxes 150 tons, or five and a half days at 6 lbs. per horse per hour.

Screw to be two-threaded, 15 feet 6 inches diameter. The boiler to be tubular 19 feet 9 inches long over all, 24 feet wide and 13 feet, high, and to be confined, as well as the engines, entirely below the orlop deck.

In each case the screw will be hung upon a separate shaft, capable of being easily disengaged from the driving shaft, and suitable means will be provided for shipping and unshipping it through a trunk extending from the upper part of the aperture for the screw, to the upper deck.

A fan blower, with all the necessary apparatus, will be provided for blowing the fires in the event of injury to the funnel; and a small engine will be fitted capable of working the blower, as well as one of the pumps for feeding the boilers.

Some 42 gun frigates in the Service have been fitted with auxiliary power of 350 horses, the screw being driven direct, at a velocity of 55 revolutions per minute, the diameter of screw being 13 feet 6 inches, the pitch 15 feet and the length 2 feet 6 inches.

The Government require the usual list of spare gear as supplied for paddle wheel engines, with the addition of a spare propeller and outside shaft complete.

Experiments with "Dwarf."—We subjoin a table of experiments with the "Dwarf" screw steam tender, which show the effect produced on the speed of the vessel by variations in the pitch, the length, the area and the speed of the screw.

EXPERIMENTS WITH H.M. SCREW STEAM TENDER "DWARF."

Draught of water, (constant,) aft, 7 feet; forward, 5 ft. 10 in. Length between the perpendiculars, 130 feet. Breadth, 16 feet. Depth in hold, 9 feet. Burthen in tons, 163$\frac{32}{94}$. Engines, diam. cylinder, 40 in. Length of stroke, 2 ft. 8 in. Nominal horse power, 90.

The Screw is two-threaded, with common pitch.

No. of Experiment.	Date of Experiment.	Diameter of Screw.	Pitch of the Screw.	Length of the Screw.		Area of the Screw.	Angle of the Screw.	Average revolutions of the engine per minute.	Average revolutions of the Screw per minute.	Proportion of revolutions of the Screw to the Engine.	Average Pressure of steam in the cylinders, taken by the indicator.	Average Pressure on spring balance of dynamometer.	Average Pressure at the end of the shaft, indicated by the dynamometer.	Speed of Screw per hour.	Slip of Screw per cent.	Speed of Vessel per hour.	Gross average power exerted by the Engine in the Cylinders.	Speed of Vessel computed, if the power exerted had been 160 horses.	Gross horses power in the Cylinder computed as necessary to arrive at a speed of nine knots per hour.	Average power expended in propelling Ship as indicated by dynamometer.	Ratio of gross indicator to dynamometer power.	Difference between indicator and dynamometer power.
	1845.	ft. in.	Feet.	ft.	in.	Feet.					lbs.	lbs.	Tons.	Knots.		Knots.	Horses.	Knots.		Horses.		Horses.
1	June 5	5 8	8 0	2	6	D = 22·2	24° 12′	28·333	146·165	5·16:1	11·36	33·87	1·361	11·532	25·017	3·647	130·736	9·249	147·41	80·993	1·62	49·743
2	... 7	...	...	2	0	C = 17·8	...	29·604	152·759	...	12·6	37·191	1·494	12·052	25·697	3·955	151·512	9·119	153·807	92·102	1·65	59·41
3	... 11	...	...	1	6	B = 13·3	...	30·124	155·441	...	11·2	35·3	1·418	12·264	27·112	3·939	137·043	9·413	139·871	87·263	1·57	49·78
4	.. 13	...	...	1	0	A = 8·9	...	32·166	165·973	...	12·92	37·728	1·516	13·094	30·419	9·111	168·805	8·95	162·71	95·06	1·78	73·745
5	... 30	...	10·32	1	$8\frac{7}{16}$	B = 13·3	30° 6′	31·763	127·050	4 : 1	11·53	26·697	1·073	12·931	33·447	8·606	148·757	8·818	170·138	63·538	2·33	85·219
6	July 2	...	...	2	$3\frac{1}{4}$	C = 17·8	...	30·823	123·293	...	10·89	23·807	·957	12·548	30·823	8·743	136·342	9·222	148·722	57·561	2·34	78·781
7	... 4	...	...	1	$1\frac{5}{8}$	A = 8·9	...	34·943	139·772	...	10·85	25·	1·004	14·225	36·387	9·049	153·998	9·164	151·51	62·561	2·44	91·437
8	... 8	...	...	2	$10\frac{1}{16}$	D = 22·2	...	30·899	123·595	...	11·46	22·75	·914	12·572	29·295	8·889	143·832	9·21	149·003	55·924	2·57	87·908
9	... 16	...	13·23	1	$10\frac{9}{16}$	B = 13·3	36° 37′	36·489	114·213	3·13:1	10·91	29·25	1·175	14·902	40·411	3·88	161·701	8·849	168·345	71·83	2·25	89·871
10	... 18	...	...	3	$1\frac{5}{8}$	D = 22·2	...	34·048	106·571	...	10·81	33·687	1·353	13·905	38·706	3·523	149·501	8·718	176·033	79·4	1·9	70·101
11	... 19	...	...	1	$3\frac{1}{16}$	A = 8·9	...	40·222	125·894	...	10·83	43·25	1·738	16·426	44·758	9·074	176·937	8·775	172·643	108·53	1·62	68·407
12	... 22	...	...	2	6	C = 17·8	...	35·714	111·784	...	11·47	43·385	1·743	14·585	39·479	8·827	166·39	8·713	176·366	105·906	1·57	60·484
13	... 29	...	...	2	6	C = 17·8	...	26· 20	105·682	4 : 1	14·10	44·319	1·781	13·789	39·88	8·29	151·957	8·434	194·439	101·604	1·49	50·353
14	Aug. 1	...	...	3	$1\frac{5}{8}$	D = 22·2	...	25·397	101·588	...	13 53	43·069	1·73	13·524	39·565	8·01	139·574	8·517	197·986	95·403	1·47	44·171
15	... 5	...	...	1	$10\frac{9}{16}$	B = 13·3	...	26·028	104·111	...	11·09	42·	1·688	13·584	41·674	7·923	117·246	8·788	171·853	92·025	1·27	25·041
16	... 6	...	...	1	$3\frac{1}{16}$	A = 8·9	...	29·830	119·322	...	12·17	48·111	1·933	15·568	46·172	8·38	147·458	8·611	182·669	110·586	1·32	36·872
17	... 13	...	...	1	$3\frac{1}{16}$	A = 8·9	...	21·658	111·756	5·16:1	13·26	40·888	1·643	14·581	46·122	7·856	116·651	8·729	175·426	88·831	1·31	27·82
18	... 15	...	...	1	$10\frac{9}{16}$	B = 13·3	...	19·889	102·629	...	14·14	40·416	1·624	13·39	42·913	7·644	114·232	8·553	186·466	85·436	1·34	28·796
19	... 16	...	...	2	6	C = 17·8	...	18·980	97·938	...	14·46	43·083	1·731	12·778	41·165	7·518	111·479	8·48	191·255	89·573	1·23	21·906
20	... 18	...	...	3	$1\frac{5}{8}$	D = 22·2	...	18·491	95·413	...	14·49	42·555	1·71	12·449	39·553	7·525	108·831	8·556	186·191	88·557	1·22	20·274
21	... 19	...	10·32	2	$10\frac{1}{16}$	D = 22·2	30° 6′	22·514	116·171	...	13·92	49·486	1·988	11·823	29·993	8·284	127·297	8·94	163·24	113·363	1·12	13·929
22	... 20	...	...	1	$8\frac{7}{16}$	B = 13·3	...	24·880	128·382	...	13·26	47·566	1·911	13·066	36·423	8·397	134·005	8·812	170·418	109.272	1·23	24·733
23	... 21	...	...	1	$1\frac{5}{8}$	A = 8·9	...	26·656	137·546	...	12·79	49·622	1·994	13·999	38·51	8·608	138·481	9·033	158·272	113·125	1·17	20·356
24	... 25	...	...	2	$3\frac{1}{4}$	C = 17·8	...	23·864	123·136	...	13·06	48·347	1·943	12·532	32·948	8·403	126·594	9·085	155·538	112·349	1·13	14·215
25	... 28	4 $5\frac{1}{2}$	...	1	$8\frac{7}{16}$	B = 13·3	24° 12′	30·807	158·965	...	11·58	45·069	1·811	15 677	49·353	7·94	144·905	8·222	211·033	98·961	1·40	45·944

Diameter of the Screw.—The diameter of the screw should, in all cases, be made as large as the draught of water will admit; and for navigation in smooth water, its upper edge need not be kept more than a few inches below the surface. For sea-going vessels, it is advisable to keep it eighteen inches or two feet below the average surface of the water;

otherwise it will lose much of its power by the pitching motion.

The Area of the Screw.—By the *area* of the screw is understood the plane projection of the resisting surface. On examining the "Dwarf's" experiments, it will be seen that the speed of the vessel gradually increased a little as the length of the screw was reduced from 2 feet 6 in. to 1 foot; the area corresponding to each length being respectively 22·2 and 8·96 feet.

It appears at first sight remarkable, that so great a variation in the resisting surface should cause so little disturbance in the speed of the engines, or of the vessel itself; and it shows very clearly how small a segment of the pitch is requisite to absorb all the power which the reaction of the water is capable of imparting. The fact is, that the water in which the screw revolves acquires very soon its rotatory motion; and, as its resisting power is thus destroyed by the action of the *leading* part of the thread only, any additional length of screw, after the power has been absorbed, only retards by friction. The experiments with the "Rattler" were commenced with a screw 5 feet 9 inches long, which was gradually shortened, with an improved speed in the vessel, until it arrived at its present length of 15 inches.

The Pitch of the Screw.—The proper *pitch* to be given to the threads of the screw, or in other words, the space through which the screw would advance during one revolution if working in a solid, must of course be proportioned to the estimated speed of the propeller shaft, taking the probable *slip* of the screw into account.

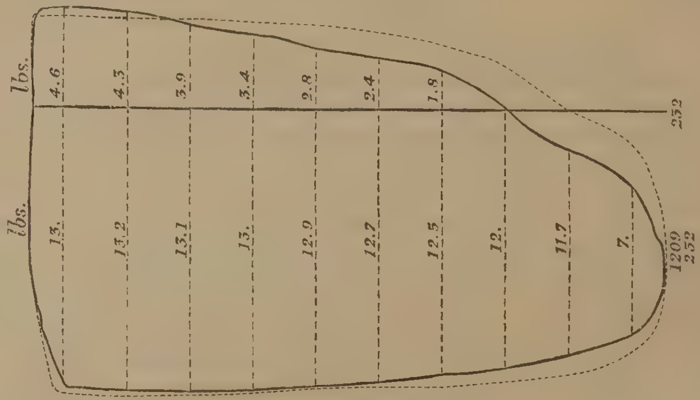
The Slip.—What is technically called "slip" is the excess of the velocity of the screw (calculated by multiplying the pitch into the number of revolutions in any given time) over the actual velocity of the vessel, and is a necessary consequence of the yielding nature of the substance through which the screw advances; for it is evident that a screw working in a solid has no slip, its progression in such a case being exactly equal to the pitch multiplied by the number of revolutions. The diameter of the screw, and the speed at which it is driven, modify, in a great measure, the amount of slip, which diminishes in proportion as the diameter and velocity of the screw are increased; as is apparent from the "Dwarf's" experiments.

The diameter of the "Rattler's" screw is 10 feet, and the average slip only 15 per cent.; while the "Dwarf" and "Fairy," with a screw between 5 and 6 feet diameter, show an average slip of about 35 per cent. The slip is also affected by the form of the vessel's run, and the manner in which the water closes in upon the screw, so that it is necessarily very uncertain, and will probably vary in every ship.

Increasing Pitch.—An ingenious modification of the screw has been patented by Mr. Woodcroft, who forms the thread of his propeller with an *increasing pitch* from the fore part towards the after part. The effect of this is, that after the leading portion of the thread of the screw has communicated its own retrograde motion to the water immediately in contact with it, (which is thus rendered incapable of further reaction upon a screw revolving with the same velocity as itself), a fresh impulse is obtained by causing the tail of the screw, as it were, to revolve with an increased velocity, the screw thus continuing to find a resisting medium in the water from the beginning to the end of the segment in revolution. Although excellent in theory, this plan cannot be said to have shewn any appreciable advantage in practice; owing, probably, to the smallness of the segment of the whole pitch which is necessary to absorb the power of the engines, and which can hardly afford a field large enough to make this refinement of much value.

The "Rattler" and "Fairy."—We may observe that the best experiments with the "Rattler" were made with the common two-threaded screw, 10 feet diameter, 15 inches long, and 11 feet pitch, when the speed of the engines was 26·19 revolutions per minute, and of the screw, 103·67; the speed of the vessel being 10·074 knots, and the slip of the screw only 10·42 per cent. Fig. 13 represents two indicator diagrams, taken from the "Rattler's" engines at different speeds, viz., 27 revolutions, as shown in black, and 18·1 revolutions, as shown in dotted lines. The "Rattler" has certainly proved a successful vessel for a screw steamer, (although her gearing cannot be much depended upon), and she is a particularly good sailer under canvas. The "Fairy" also acquits herself well under favourable circumstances, when she will achieve about twelve knots an hour. Her dimensions are as follows:—

Fig 13.



The Rattler's after Engine.

Length between the perpendiculars	144 ft. 8 in.
Breadth of beam for tonnage	21 ft. 1½ in.
Depth in hold	9 ft. 10 in.
Burthen in tons, (builders' old measurement,)	312
Horse power	130
Draught of water	6 feet.
Diameter of screw	5 —
Length of screw	15 inches.
Revolutions of engines per minute	48
— screw per minute	240

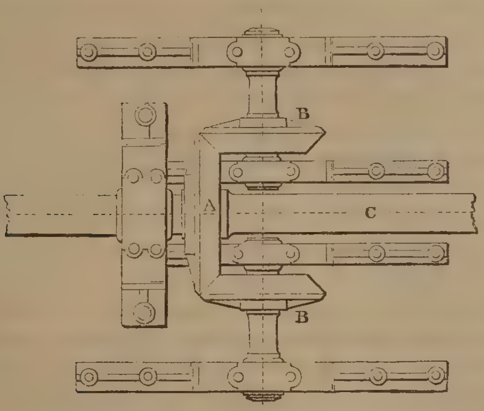
She has a round bottom and very fine extremities.

The "Rattler" is 888 tons, builder's old measurement, and has 200 nominal horse power. The submerged area of the midship section, (at 12 feet 3 inches draught) is 380 square feet, and the effective surface of the best screw was 23 square feet.

The "Great Britain."—Although the results obtained from the screw in the "Great Britain" were decidedly good, so far as they went, we can hardly judge of their value, as compared with corresponding results from the paddle wheels, owing to the extreme singularity of the case. It is well known, however, that the effective power of the engines increases in a much higher ratio than simply in proportion to the tonnage of the vessel; that is to say, that engines of 1000 horse power would propel a vessel of 3000 tons much faster than engines of 500 horse power would propel a vessel of 1500 tons. This is apparent from the consideration that the tonnage increases as the cube of the dimensions, while the resisting surface of the vessel increases simply as the dimensions; and this fact is amply attested by the performance of our largest class of steamers, which are usually at the same time the fastest, although not built so much with a view to speed as many of their smaller and slower brethren.

Thrust of the Screw.—The thrust exerted by the screw in propelling the vessel is usually received on a steel plate, near the fore bulk-head of the engine room, against which a steel pin in the end of the screw shaft presses, the point of contact being kept cool by a stream of cold water, or otherwise. But for some such provision, the heat generated by friction is so great as to have actually welded the steel pin on to the plate, and, in any case, the wear between the surfaces is very rapid. To obviate this, an arrangement of friction cones, such as that represented in Fig. 14, has been proposed for receiving the thrust of the screw, but we are not aware that it has been yet adopted in practice. A, is a smooth cone keyed upon the screw shaft C. B, B, are two other cones revolving on cross shafts between the standards, which transmit the thrust to the hull of the vessel. A friction plate must be provided on the face of the outer stern post for receiving the thrust of the screw in going astern.

Fig 14.



Surface Cones, for receiving the thrust of the Screw Propeller

Chap. II.—Marine Boilers.

Average Consumption of Fuel.—The evaporative power of coal in marine boilers of good construction may be estimated at 9 lbs. of water evaporated by 1 lb. of coal. The rate at which combustion proceeds may be assumed, (as an average), at 13 lbs. of coal per square foot of grate bar surface per hour. If stated in terms of the nominal horse power, 10 lbs. of coal, per horse per hour, may be taken as the average consumption of sea-going steamers.

As the horse power of an engine is such an indefinite term, depending upon the pressure of the steam and the velocity of the piston, it has been assumed in the following calculations to equal one-half of the gross horse power in the cylinders, as shewn by the indicator. We have also to remark, that should the boilers be expected to require more fuel than 10 lb. per horse per hour, for the efficient supply of steam to the engines, the dimensions given below must be proportionably increased, or if less, diminished. While the engines are working expansively, it is usual to allow for a consumption of 6 to 8 lbs. of coal per horse per hour; in which case, the performance of one, and sometimes in large engines, of two boilers may be temporarily dispensed with. It is by no means an unfrequent case, on the other hand, for a fast steamer, using tubular boilers, and steam at 15 lbs. pressure, with little or no expansion, to consume 14 or 15 lbs. of coal per nominal horse power per hour.

Proportions of Marine Boilers.—In the following proportions for marine boilers, it is believed that a sufficient economy of combustion will be afforded without unduly increasing the weight and cubical dimensions of the boilers, which are both considerations of the utmost importance, and which have now led to the almost universal abandonment of "flue" for "tubular" boilers.

In tubular boilers, the whole heating surface of the tubes should be calculated at 14 square feet per nominal horse power, without making any deduction for ineffective surface, and the *effective* heating surface of the furnaces and up takes should be about 2·75 square feet additional, making in all, 16·75 square feet of heating surface; or allowing one-third for ineffective surface in the bottoms of the tubes, we have about 12 square feet per horse power of "effective" surface either for flue or tubular boilers. The grate bar surface in boilers of considerable size, should be at least 80 square inches per horse power, but in smaller boilers from 90 to 100 square inches should be allowed. The area of the first flue, or the clear area through the tubes, should equal one-sixth of the grate bar surface led into them. The combined area over the bridges should be equal to the area of the tubes or first flue. It is preferable to fix the tubes in the tube plates by countersinking the holes, and hammering the ends of the tubes into them, in place of using ferrules, which contract the area through the tubes considerably.

Chimney.—The area of the flues in a flue boiler may be gradually diminished from the furnace to the chimney, which should have an area of three-fourths of the first flue, or flues combined. No rule can be given as to the height of the chimney in a steam vessel, as this is limited simply by its appearance in proportion to the vessel and the masts, and by its effect upon the vessel's stability. It is found that funnels of large diameter create a much quicker draft than small ones, *ceteris paribus*, owing to the more rapid absorption of the heat from the small funnel, which presents a proportionably larger surface to the external air. It is usual to inclose the bottom part of the funnel in single or double casings of thin iron, to check this injurious radiation or convection of the heat.

Furnaces.—The furnaces or "fire boxes" should be about 3 feet 6 inches deep, so as to allow of a roomy ash-pit under the furnace doors, without which a good draft will not be obtained. The doors may be 10 to 12 inches high, and of a width proportioned to that of the furnaces, say, 20 to 22 inches. The front of the grate bars should be at a height of 30 inches above the firing stage, and should slope down towards the bridge, with an inclination of about 2 inches to the foot. They are best formed of wrought-iron, in lengths of about 30 inches, to suit the length of the furnace, which should not exceed 6 feet 6 inches or 7 feet, if possible; this being the utmost length which can be advantageously fired. The bars should be made from five-eighths to three quarters of an inch thick on the upper edge, about 3 inches deep, and with $\frac{7}{16}$ inch air spaces between each.

Steam Capacity.—The capacity for steam over the water in the boiler should be at least 3 cubic feet per horse power, and may be made nearly double this amount with advantage, where height in the boiler is not objectionable. This is a point of much importance, since, if sufficient steam room is not provided, the boiler will certainly *prime*, owing to the unequal and violent ebullition which then ensues, as the pressure varies in the boiler at each withdrawal of a cylinderful of steam. The steam should be taken from the boiler at a height of not less than five feet above the surface of the water, but this will be of no avail to prevent priming should the steam room be insufficient.

The Flues.—From 10 to 12 inches of water should be kept over the tubes or flues, and 14 or 15 inches over the furnaces. The flues of a flue boiler are usually made to increase slightly in height as they recede from the furnaces, so that the depth of water over them may gradually diminish, (say from 15 to 10 inches,) as the contained air becomes cooler. For the same reason, and perhaps to facilitate the draft a little, the tubes of tubular boilers usually slope 3 or 4 inches in their length towards the funnel. The flues should be of such a dimension as will allow of a boy getting readily through them to clean out the deposit, say 4 feet high, by 16 inches wide, but care must at the same time be taken that their area is not more than just sufficient for the products of combustion to pass readily through. Should they be too roomy, or, which is worse, should their area be unduly contracted or enlarged at any one place, a deposit of soot and small cinders will be the consequence, which will impair the heating surface, and very probably destroy the plates by corrosion if any water gain access by a leak in the flues.

The Water Spaces.—The water spaces between the furnaces should not be less than 5 to 6 inches wide, and between the flues 4 to 5 inches. The spaces over the crowns of the furnaces should be not less than 10 inches, so as to allow of a man-hole between the arched tops. The bottom water spaces should be 8 inches, for the convenience of cleaning. It is usual to allow a space of 1 inch between the tubes of a tubular boiler, and these are best arranged in perpendicular rows, one over the other; by which means the steam escapes from their surface more readily than when arranged zig-zag. When water bridges are used, they should not be less than 10 to 12 inches wide, and their tops should incline about 3 inches to the foot, to allow of the steam escaping more readily from the interior surface.

Heating Surface arranged to permit the easy Escape of the Steam.—Care must be taken, in proportioning the boiler, that no part of the heating surface may be so situated that the steam will not readily escape from it to the surface of the water, as in such case the plate, being left in contact with steam in place of water, becomes intensely heated, and destroyed, and an explosion is not an unfrequent result. It is found in practice that perpendicular heating surface, such as the sides of the rectangular flues in a flue boiler, is by no means so efficient for raising steam as an equal amount of horizontal surface, such as the tops of the same flues. The reason of this is sufficiently obvious; as the steam, in the first case, rising perpendicularly from each portion of the surface of the metal, forms a film or stratum of vapour in contact with the sides of the flue, thus preventing the free access of the water to the hot metal; but, in the other case, the steam, leaving the iron as soon as generated, allows the water to be constantly in contact with it. From the same cause, the flat bottom of an internal metal flue is very inefficient as heating surface; and it is also found that plates which are liable to be exposed in this manner to imperfect contact with the water, either as side or bottom surface, are subject to a much quicker wear than the tops of the flues of the same boilers. The plan of covering the bottoms of the flues with a non-conducting substance, as bricks or cement, has been adopted without any appreciable diminution in the evaporative power of the boiler; and is believed by some to have increased the durability of those plates, which are generally amongst the first to give way. It may be objected to this plan that, owing to the very unequal degrees of expansion by heat between the iron and the non-conducting lining, it is impossible to maintain an unbroken joint between their surfaces, so that a space would be thus formed in which brine might collect in contact with the plates, and do more mischief than the original disease. It is also of much importance that the surface of the plates should be exposed for examination as much as possible.

Strength of the Plates.—The pressure at which flue boilers, (supplying condensing engines,) are usually worked in this country is 5 lbs. to the square inch, and tubular boilers at 10 lbs. to 15 lbs. per square inch. A practical limit is imposed to increasing the thickness of the plates of an iron boiler, by the defective conducting power of the material; and it not unfrequently happens that boilers are weakened and destroyed by the very means adopted for securing additional strength, viz. by giving an injudicious thickness to those plates exposed to the greatest heat.

It is now generally understood that if the plates of the furnaces are made thicker than about three-eighths of an inch, they are liable to be warped and "burnt" by the action of the flame, in consequence of the conducting power of the metal being insufficient to transmit the heat of the flame through the plate with such rapidity as to prevent the exposed side of the plate being softened and weakened by the intense heat, even although the other side of the plate may be in perfect contact with the water. From this cause, lapped joints, or rows of rivet heads, should be avoided as much as possible in parts of the boiler exposed to *flame*, which is much hotter than *radiated heat*. The best position for the joints of the plates in the fire boxes is in a slanting line, just below the fire bars, in the ash-pit; and the plates for this part should be made as large as it is possible to have them rolled sound.

The usual thickness of iron plates for boilers is as follows: for fire boxes, $\frac{3}{8}$ in. Lowmoor; flues, sides and tops, $\frac{5}{16}$, (the tops are sometimes $\frac{1}{4}$ in.,) bottoms, $\frac{3}{8}$ and sometimes $\frac{7}{16}$ in. Outside shell of boiler, $\frac{3}{8}$ in. throughout, although the bottom is sometimes made $\frac{7}{16}$ in.; the up-takes to the bottom of funnel should be $\frac{3}{8}$ in. The tube plates in tubular boilers are made from $\frac{1}{2}$ to $\frac{5}{8}$ in., charcoal iron. The funnel may be made with $\frac{1}{4}$ in. plate round the bottom, then $\frac{3}{16}$ in. and $\frac{1}{8}$ in. at top. It should be divided by plates at the bottom, for 6 or 8 feet, into as many divisions as there are distinct boilers led into it.

Division of the Boilers.—We may observe that in the case of engines of large power it is preferable to divide the boilers into several distinct pieces, each complete in itself, and capable of being used either in conjunction with the others or separately. This affords facilities for examining, repairing if necessary, and cleaning the boilers in succession during a long voyage, and also for economizing fuel by the disuse of one or even two of the sections. Boilers of great weight and size are also very inconvenient for lifting into and out of the vessel, and require the deck to be framed with boiler hatches of a proportionally large size. They are, therefore, generally divided into 2, 3, 4 or 6 sections, according to the power, the largest pieces seldom exceeding 120 or 130 horse power, which will weigh about 15 tons, for a tubular boiler.

We give, in the Appendix, a table of the proportions of marine boilers by our best makers.

Staying of Boilers.—It is usual, on the Thames, to stay the flat portions of tubular boilers with about $1\frac{1}{8}$ inch round rods, from 16 to 18 inches apart each way, and flue boilers not quite so heavily. It is highly injurious to stay weak plates at long distances, as the alternate distension and contraction of the plate between the stays causes it to buckle round each stay every time that the pressure of steam is added or removed. This action, in time, wears a furrow round the fastening of the stay by throwing off the scale from the surface of the plate and opening the fibre of the iron, the circular piece of plate to which the stay rod is attached remaining at the same time quite sound. We believe this curious effect has been sometimes attributed to the agency of galvanism, that convenient scapegoat which has proved so useful in explaining the mysteries of engineering.

Galvanic Action.—It is very necessary, however, in designing marine engines and boilers, to guard against the destructive effects of galvanic action in all cases where two metals of different degrees of solubility in salt water, (such as iron and brass,) are placed in juxta-position; when the iron, being the most oxidable metal, suffers a rapid corrosion. To prevent this action, as well as for economy, it is now usual to furnish marine boilers with iron tubes in place of brass; from $2\frac{1}{2}$ to $3\frac{1}{4}$ inches external diameter, and about $\frac{3}{32}$ to $\frac{1}{8}$ inch thick.

A hard, non-bituminous coal is preferable for tubular boilers, and the tubes should be cleaned out with brushes as often as convenient.

Clothing of Boilers.—Although it must be allowed that in all cases the clothing of marine boilers with non-conducting substances, such as hair-felt, wood, &c., is highly advantageous for the production of steam, yet this practice is alleged in some instances to have induced a rapid wear in the plates of the boiler. This unlooked-for result is most apparent in boilers which are frequently used and disused alternately; the corrosion taking place on the interior surface. The only conjecture we can offer as to its cause is, that owing to the alternate wetting and drying of the plates of a *clothed* boiler, the rust may be more apt to scale off, and thus constantly present a clean surface for corrosion, this action recurring each time that the water is blown out of the boiler; but when, on the other hand, the boiler is *naked*, the internal surface never thoroughly dries, owing to the evaporation being checked by the low temperature, and the saturation of the confined air. The clothing of marine boilers which make long voyages is never attended with these injurious results, nor are they ever experienced in land boilers, that we are aware of. Clothing Her Majesty's steam boilers is a most complicated affair, as will be seen by referring to the specification at page 34, and is only effected after a vast expenditure of hair-felt, canvass, wood, iron straps, paint, sheet lead, bricks and cement.

Bedding of Boilers.—The manner of *bedding* marine boilers is a point of much importance, and will materially affect the durability of the bottom plates. The practice with the Admiralty, as well as with some private engineers, is to form a close platform of two-inch fir deals over the keelsons, upon which the boiler is then bedded with a cement or mastic of lime and drying oil, laid about one inch thick over the timber. This sets quite hard, and prevents the bilge water, (which in wooden ships is highly acid,) from washing up to, and corroding, the plates of the bottom. The cement is also intended to stop any leaks which may break out in the bottom of the boiler, as well as to strengthen it in the event of a rapid corrosion taking place inside. Unfortunately, a perfectly close contact cannot be maintained between the iron and the cement, from the unequal degree of expansion of the two materials by heat, so that the effect of a leak in the bottom plates is frequently that the brine extends itself for a large space between the two surfaces, and the wear of the boiler is increased in place of diminished. The best practice is, perhaps, that of resting the boiler on saddles of cast-iron fixed on the boiler bearers, which leaves the bottom exposed for examination, painting, and small repairs if necessary, the bottom of the vessel under the boilers being, at the same time, kept clean and dry by the bilge pumps.

Duration of Iron Boilers.—As will be easily understood from the preceding remarks, the duration of an iron boiler varies according to the treatment it has received, and the facilities afforded by its construction for thoroughly examining, cleaning and repairing the parts. Thus, while some have been worn out with three years' service, others of the same thickness of plate have lasted six and eight years. A Government steamer is seldom in commission longer than four years at one time, and at the end of this period the boilers very frequently require renewal; or, if they are not much worn, and if the vessel be of such a class that she will probably not be sent on foreign service, the boilers are repaired and made to serve other two, or it may be, four years.

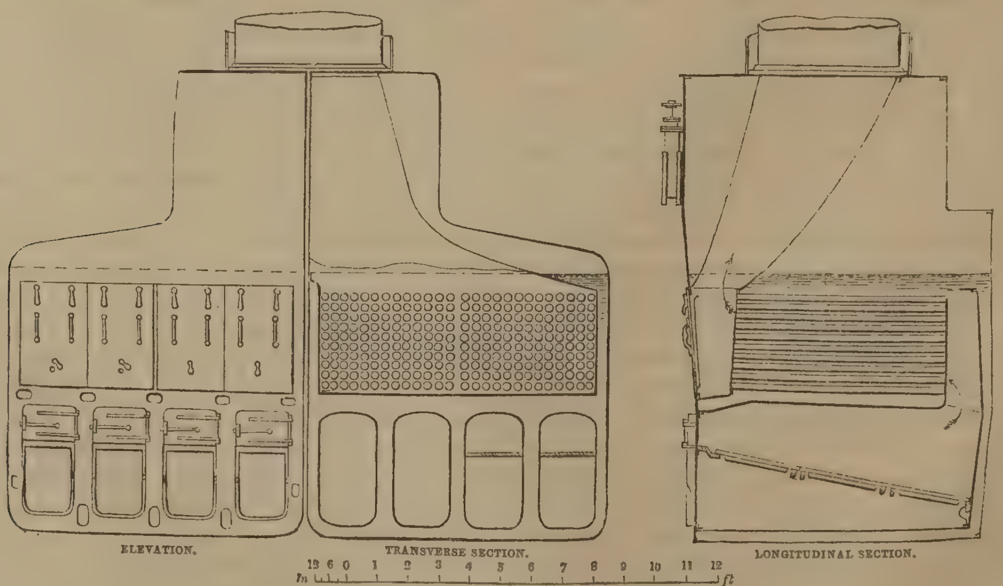
Copper Boilers.—Although the durability of copper boilers compared with iron may be stated as 4 to 1, their greatly superior first cost has hitherto proved a bar to their introduction, except under peculiar circumstances, as where the frequent renewal of iron boilers would be attended with inconvenience. Their cost is about five times that of iron boilers of the same weight, and copper is only three-fifths of the strength of iron when cold, or less than half the strength at 500°. Iron, on the contrary, increases in strength from 32° to 550°, when its strength is a maximum, being then sixteen per cent. greater than when cold. It must be observed, however, that although a copper boiler is not so strong as a new iron one of equal weight, it retains its original strength almost entire for a long time, whilst the other is every day getting weaker by corrosion. Hence, an iron boiler, when new, must have a great superabundance of strength, in order to allow for the wearing of the plates by use. Owing to its superior conducting power, viz., as 2·4 : 1, the copper marine boiler will generate an equal amount of steam with, perhaps, one-fifth less heating surface than an iron one, but it would be advisable to distribute the weight of material thus gained in making the plates a little thicker than if for an iron boiler. When a copper boiler is worn out, the old material will fetch a good price, whereas an old iron boiler is almost worthless—the price obtained for the scrap iron being very little more than the cost for breaking it up.

Practical Rules for Proportioning Boilers.—The following is the mode of proportioning marine boilers adopted by the engineer to our largest and most successful commercial steam company. "I first find the quantity of coal which the boiler will probably burn, calculating this at 8, 10 or 12 lbs. per horse per hour, according to what I know, or expect, of the engines. I allow for combustion to be carried on at the rate of 13 lbs. per square foot of grate bar surface per hour, and I thus find the total area of the grates by dividing the whole quantity of coal burnt per hour, in pounds, by 13. For 'effective' heating surface, I allow 18 square feet for each square foot of grate bar, but 16 or 17 square feet will do, if pinched. In calculating tube surfaces I always deduct one-third for bottom surface, and therefore non-efficient. For the area through the flues or tubes, I allow 2 square inches for each pound of coal burnt per hour. For the area of the chimney, I allow two-thirds of the area of the flue as found above. In common flue boilers I prefer reducing the area of the flues gradually and regularly as they recede from the furnace, in place of suddenly contracting them at the foot of the chimney. The area over the bridges I make the same as the area of the first flue." These calculations, if stated in terms of the nominal horse power, will stand thus, assuming 8 lbs. of coal per horse per hour as the consumption of our engines. The grate bar surface will then be 88.5 square inches per horse power, and the area of the first flue, (or over the bridge,) will be $\frac{1}{5\cdot58}$ of the grate bar surface. The "effective" heating surface per horse power will be 11·1 square feet, one-third of the tube surface being deducted as inefficient.

Experimental Boiler, H. M. Dockyard, Woolwich.—The following table contains the results of some experiments, conducted at Woolwich dockyard, with an eighty horse tubular marine boiler, of the ordinary construction, such as that represented in the wood cuts, Figs. 15, 16, and 17; which show a front elevation and two sectional views of a pair of boilers, by Messrs. Penn, of 260 horse power collectively.

The extreme measurements of the experimental boiler are 9 feet long, by 8 feet 6 inches wide; thus covering an area of 76½ square feet in the vessel, or ·956 square feet per horse

Fig. 15. Fig. 16. Fig 17.



Tubular Marine Boiler, as constructed in H. M. Dockyard at Woolwich.

power. It has three fires, each 8 feet by 2 feet 2 inches; which thus afford 93·6 square inches of grate surface per horse power. The total "effective" heating surface, after deducting one-third for the bottoms of tubes, is 13 sq. feet per horse power. There are 176 iron tubes, (arranged 22 horizontally, and 8 vertically,) each 3 inches outside diameter and 6 feet long. The furnaces are 2 feet 10 inches high, and the boiler itself is 12 feet 2 inches high. The capacity for steam is 167 cubic feet; and the total space occupied by the boiler is 761 cubic feet.

EXPERIMENTS WITH A TUBULAR MARINE BOILER. H. M. DOCKYARD, WOOLWICH.

Number of Experiment.	Duration of Experiment.	Number of Fires.	Dimensions of the Furnaces.	Total Surface of Furnace Grates.	Bar Surface per cub. ft. evaporated per hr.	Kinds of Coals used in the Experiments.	Quantity of Coal used.	Quantity of Water evaporated.	Ratio of effect at temperature of the Feed Water.	Ratio of effect with the Feed water at 100 degrees.	Evaporation per hour in cubic feet with the Feed Water at 100°.	Ratio of effect in proportion to the Grate Surface.	Coals burnt per sq. ft. of the Fire Grate during the Evaporation.	Coals burnt per sq. foot per hour.	Temp. of the Water supplied to the boiler.	Total evaporating surface in the boiler in square feet.	Evaporating surface for each cubic foot of water evaporated per hour.
	H. M.		Ft. In. Ft. In.	Sq. Ft.	Sq. Ft.	Welsh Coal.	Lbs.	Lbs.			Cub. Feet.		Lbs.	Lbs.	Deg.	Sq. Ft.	Feet.
1	6 30	3	8 0×2 2	52	·515	Leur's Merthyr	4648	41040	8·83	9·183	105·063	8·83:1	89·38	13·75	56	1057	10·06
2	5 50	—	—	—	·489	Wood's —	4704	38760	8·24	8·584	110·759	8·384:1	90·46	15·51	54	—	9·54
3	*8 6	—	—	—	·825	— —	3808	31920	8·382	8·733	65·689	14·145:1	73·23	9·04	—	—	—
4	2 55	—	5 3×2 2	34·125	·546	— —	1176	11400	9·694	10·108	65·209	9·362	34·46	11·81	53	1057	16·22
5	6 20	—	—	—	·658	Leur's —	1904	20520	10·777	11·228	54·008	11·282	55·77	8·81	54	—	19·57
6	6 24	—	—	—	·499	— —	2576	27360	10·621	11·075	71·322	8·556	75·49	11·79	53	—	14·82
7	5 55	—	4 0×2 2	26	·351	Handpicked —	2632	27360	10·395	10·868	77·351	6·018	101·23	17·11	50	—	13·66
8	5 58	—	8 0×2 2	52	·773	— —	2352	25080	10·663	11·177	70·494	13·254	45·23	7·58	47	—	14·99
9	4 10	—	4 0×2 2	26	·495	— —	1120	13680	12·214	12·802	55·062	8·487	43·08	10·34	—	—	19·19

Average amount of water evaporated by one pound of coal during nine experiments, 9·35 lbs.
* Common Flue Boiler.

These results indicate the relative economy of slow and rapid combustion on the bars, and the effect produced on the evaporative power of the boiler by diminishing the area of the grate bar surface.

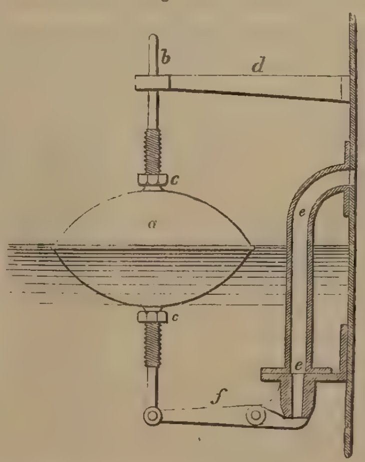
Wood Fuel.—When wood is burnt in marine boilers, 2½ lbs. may be considered equivalent to 1 lb. of coal, on the average. We understand that the boats on the Swiss lakes consume about 22 lbs. of wood per horse per hour. The furnace should of course be made very large to hold the fuel, and the flues very roomy to accommodate the gases.

Turf Fuel.—Peat has been occasionally used in Ireland as fuel for steamers, though not hitherto with very beneficial results. Its calorific value depends very much upon the degree of dryness to which it has been brought, and it has been found by experiment with a tubular marine boiler that, under the most favourable circumstances, it required 51 lbs. of good peat to convert one cubic foot of water into steam, which is about one-sixth of the duty of coal. The boiler with which this experiment was made had 6½ square feet of grate bar, and 280 square feet of total heating surface, (without any deduction). When the turf is kiln-dried, it has a strong tendency to re-absorb moisture, and should consequently be used as soon as possible after its preparation. The average cost of turf, in Ireland, is from 5d. to 7d. per box of twenty cubic feet.

Boiler Incrustations.—The fruitful cause of deterioration in marine boilers is the deposition of salt, lime, and other mineral substances from the water upon the internal surface of the plates. Such incrustations, being imperfect conductors of heat, cause the plates to be rapidly "burnt," or oxidized, by preventing contact with the water, to which the incrustations themselves are unable to transmit the heat of the fire from the plates with sufficient rapidity. Hence, the plates being unduly heated, waste away on each surface, and, becoming soft by the intensity of the heat, they are frequently warped and weakened in a very dangerous degree. Besides this, there ensues a serious loss of evaporative power in the boiler, from a portion of the heat passing off by the chimney without being absorbed by the water.

Blowing Off.—The usual, and indeed the only efficient, means hitherto adopted for preventing boiler incrustations, is by the "blowing off" into the sea of a portion of water from the boiler, at short and regular intervals, according to the degree of saturation with the mineral substances from which the deposit or "scale" is elaborated, the waste being at the same time supplied by comparatively pure water from the sea. It is found necessary in practice to blow off about half the quantity of water received into the boiler as feed—that is to say, for every two cubic feet thrown by the feed pumps into the boiler from the hot well, one cubic foot is evaporated into steam, and one is blown off. This operation is performed sometimes by brine pumps, as already explained, but more frequently by means of "blow-off cocks," communicating directly between the bottom of the boiler and the sea, which are opened by hand about once every hour. The cocks used for this purpose are frequently formed on the principle of Kingston's valves, represented at Figs. 6 and 7, (page 27), and are formed of gun-metal, the blow-off pipes being of copper a quarter of an inch thick.

Fig. 18.



Lamb's blow-off valve for marine boilers.

Lamb's Blow-off Apparatus. — A better mode of blowing off has been lately introduced by Mr. Lamb, in the boilers of the Peninsular and Oriental Steam Navigation Company, and has been already attended with much advantage. His apparatus, which is self-acting, is represented at Fig. 18, where *a* is a copper float, with a tube passing through the centre to admit the spindle *b*, which is furnished with regulating screws and nuts, *c*, *c*, for the purpose of adjusting the float to the water-line of the boiler; *d* is a bracket for guiding the spindle and keeping the float in the proper position; *e* is the blow-off pipe, which has been closed by one end of the jointed lever, *f*, on the descent of the float, as shown in the wood-cut; but as the water level rises by the admission of more feed, the same end of the jointed lever is depressed, and permits the brine to escape through the open mouth of the pipe, until the float again descends and closes the aperture. The brine thus blown off is conducted to the sea through a "refrigerator," formed by two cylinders, one enclosed within the other, the brine encircling the internal cylinder, and the feed-water passing through it. In fixing the apparatus in the boiler, the aperture of the pipe *e*, *e*, should be placed about twelve inches under the working water-line, and the buoyancy of the float must be such as will overcome the internal pressure against the mouth of the aperture. The merit of the invention lies in blowing off the supersalted water from the surface, in place of near the bottom, as is the usual practice, whereby the particles of insoluble matter, of which the scale is chiefly formed, and which are ballooned up to the surface by bubbles of steam, are caught and removed from the boiler before they have an opportunity of aggregating, and then remaining as a permanent deposit at the bottom. That the principle is correct, so far as the insoluble particles are concerned, may be easily shown by experiment, and has been already amply proved in practice. It should not be forgotten, however, that the most highly saturated brine will be found at the bottom, both from its own inherent specific gravity, in the first instance, and from the density due to the temperature of the water in that part of the boiler, where it is comparatively cool, and never agitated by boiling. Hence, it still appears desirable to blow off a certain portion of the brine from the bottom, in conjunction with Mr. Lamb's valuable apparatus.

Other Means employed for preventing Incrustations.—Various other plans for preventing boiler incrustations have been tried from time to time with the view of saving the loss of heat incurred by the system of frequent blowing off, which perhaps may amount to 4 or 5 per cent. of the whole heat generated. These are either mechanical contrivances for collecting the insoluble particles at the surface of the water, or for preventing their solidifying at the bottom by agitation with foreign substances; or else the means employed have been the introduction of a chemical solvent into the boiler with the same view. Although none of these plans can yet be safely recommended for general practice, a partial success appears to have attended Dr. Ritterbrandt's remedy of muriate of ammonia, or dilute muriatic acid, mixed with the water of the boiler. It is stated that this solvent has, in several instances, prevented the formation of hard incrustations on the plates, and has re-dissolved incrustations of carbonate of lime already formed, without injury to the material of the boiler. Its use, however, is generally believed to be attended with danger to the iron plates, and it is not effective against all kinds of deposit, as may readily be supposed.

Priming of Boilers.—The only effectual preventive against *priming* is to allow the boiler plenty of steam room, and to give it as large an area as possible at the water level. Perhaps the best mechanical contrivance which has been used to alleviate this evil is by introducing a number of wooden balls amongst the water, which, floating on the surface, facilitates the escape of the steam, and assist in breaking up and dispersing the large bubbles which cause the priming. A perforated plate is sometimes suspended at a little distance above the water line with the same view.

Boiler Mountings.—Each boiler should be fitted with at least one man-hole, and more if necessary; and a mud-hole should be placed between every two fires, at the bottom, for the convenience of cleaning; and in tubular boilers a man-hole should be cut between the tops of the fire boxes and the tubes. They should all be oval, and the doors fitted *internally*. Each separate boiler should have two safety valves, one under the control of the engineer, and the other loaded with a maximum weight, which he has no power to augment, but may relieve, as represented by Figs. 19 and 20, which exhibit two distinct modes of arrangement applicable to the same purpose. Here *a*, *a*, are the steam chests of the boilers; *b*, the safety valve chests; *c*, *c*, the safety valves, having the weights, in the one case, suspended from, and in the other, surmounting them; *d*, *d*, the rods by which the weights are relieved from the valves when necessary; *e*, the

Fig. 19.

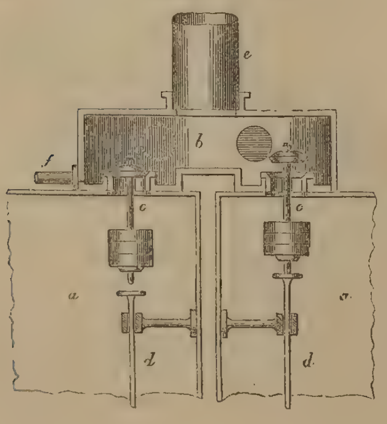


Fig. 20.

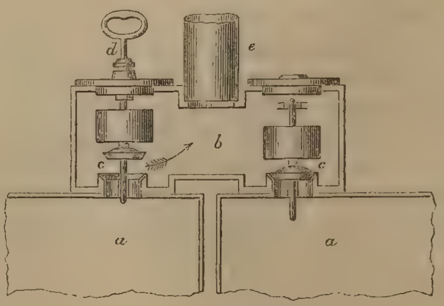
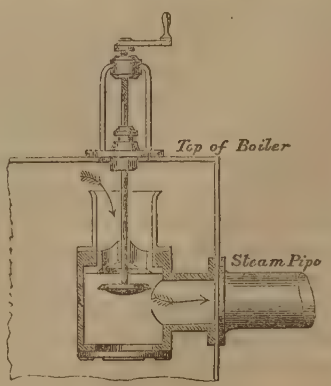


Fig. 21.



waste steam pipe; and *f*, the waste water pipe. In Fig. 19, the rods *d*, *d*, are brought within the control of a man standing on the floor of the engine room, by means of bell cranks and horizontal rods traversing the interior and passing through stuffing boxes at the front of the boilers; thus enabling the engineer to blow off the steam without going on deck, which will be necessary in the arrangement represented at Fig. 20. The usual proportion for safety valves and waste steam pipes is half a square inch of area per horse power; and for steam pipes connecting the boilers to each other or to the engines, one square inch per horse power.

Each boiler should also be provided with a stop or shut-off valve, as shewn at Fig. 21, for connecting and disconnecting; a vacuum, or reverse valve, to guard against the external pressure of the atmosphere; a glass water gauge, to show the level of the water in the boiler, and three brass water gauge cocks, with openings at proper intervals, for the same purpose; a mercurial steam gauge, to show the pressure of steam; blow-off cock and feed cock; and a small cock for the stokers, for drawing hot water to quench the cinders, &c. The vacuum valves may have one-eighth of a square inch of area per horse power. The waste steam pipe is usually formed of copper, and the top is so constructed as to prevent the escape of water which may be blown up the pipe with the steam. This is caught in a kind of trap, and conveyed back to the valve chest by a little pipe within the waste steam pipe; another small pipe leading from the valve chest down to the bilge.

Chap. III.—Stationary Engines.

I.—*The Double-Acting Condensing Engine.*

General Features.—As the general principle of the condensing stationary engine is identical with that of the marine engine already explained, it will be quite unnecessary to repeat the process of its operation, with which the reader must be already familiar. It differs only from the marine engine in employing the fly wheel and the governor to impart a certain uniformity of motion and of velocity to the piston, and thus counteract the disturbing influence of slight variations in the load of the engine. The momentum of the paddle wheels, and the practice of combining two engines to one shaft, render the use of the fly wheel unnecessary in the marine engine, and as the resistance to be overcome is subject to fluctuations so sudden and violent as to be quite beyond the control of the ordinary governor, the throttle valve is much better regulated by hand than it could be by its use.

The varieties of form which have been given to stationary engines are endless, but perhaps the most generally approved is that description known as the Long-beam engine, represented by Plates XIX. to XXII. In the example there given, it will be seen that the slide valves are formed on the long D construction, and are worked by an eccentric on the main shaft. The piston rod communicates, by means of the parallel motion links, with one extremity of the large cast iron beam, which oscillates over-head between two pedestals, supported by cast iron columns. From the other extremity of the beam, a cast-iron connecting rod, (with a fork head,) descends to the crank upon the main shaft on which the fly wheel is keyed, and which communicates the motion, by means of gearing, to the machinery. The air pump, it will be seen, is worked from a stud placed half way between the centre of the beam and the centre of the cylinder, thus having half stroke of the engine; the head of the rod being hung by a cross head from the centre of the back links for the parallel motion. The air pump and condenser are contained in an iron cistern kept full of water, (as cold as can be procured), either by the cold water pump, which is worked by the engine from a stud on the beam, or when practicable by a natural head of water. From this cistern the condenser is supplied with injection. A portion of the hot water is withdrawn from the hot well by the feed pump, (also worked from the beam,) for the supply of the boilers, and the remainder runs to waste, (if water be plentiful), or may be used in some manufacturing process; or it is sometimes cooled by exposure and evaporation in the open air, and used over again for injection. The governor is worked from a bevel wheel on the main shaft, and communicates by rods and levers with the throttle valve.

Ratio of Diameter to Stroke of the Cylinder.—Perhaps the best proportion between the diameter of the cylinder and the stroke of a stationary engine is as 1 : 2, but this may be varied considerably, to suit the circumstances of the case, without much detriment to the effect of the engine.

Expansion in Stationary Engines.—The best mode with which we are acquainted for applying the principle of expansion to stationary engines, is by Mr. Fairbairn's new plan of expansive action, represented by Plate CXXXIII. The valves, it will be seen, are on the equilibrium or Cornish principle, and are lifted at the proper time by means of graduated cams, (one to each set of valves) fixed on the heads of the vertical rods, which receive their rotatory motion from bevel wheels worked from the engine shaft. The degree of expansion is varied at pleasure by sliding, either *towards* or *from* the centre, the little rollers which receive their motion from the surface of the cams on which they press, and communicate it to the valve spindles.

The Governor.—In proportioning the governor of a steam engine, we must consider, 1st, the length of the pendulum, or the vertical distance between the point of suspension and the plane which passes through the centre of the balls revolving at their mean velocity; 2d, the range of motion; 3d, the weight; and 4th, the velocity of the balls. The proper height for the governor depends upon the following consideration, viz., that the vertical distance between the point of suspension and the plane of the revolving balls, at any given number of revolutions, is equal to the length of a pendulum oscillating in the same time that the balls make half a revolution. Thus, if we fix thirty revolutions per minute as the mean speed of our governor, the vertical height should be the same as the length of the seconds' pendulum, namely, 39·14 inches. To find the height for any other given number of revolutions, divide the constant number 375 by twice the number of revolutions. The *range* is determined by considering the greatest change of velocity which the machinery may acquire without injury, at which limit the governor should be made capable of cutting off the steam entirely.

The amount of this variation in practice is seldom permitted to exceed one-tenth of the whole velocity of the engine, in which case the range of the balls will be nearly one-fifth of the height of the point of suspension above the plane of revolution at the mean velocity; or, in the case we have supposed, 7·7 inches. To find the number of revolutions per minute, when the height is known, divide 375 by twice the square root of the height in inches, and the quotient will be the number of revolutions. The proper weight for the balls is determined by the nature of the mechanism which transmits the motion of the governor to the throttle valve; the simpler the mechanism the lighter the balls, and *vice versâ*. Hence in practice their weight varies from about 30 lbs. to 80 lbs. each, according to the friction to be overcome. Theoretically, the weight of the balls does not influence the action of the governor at all, as pendulums of equal length make the same number of oscillations in a given time, whatever be the weight of the *bob*.

The Parallel Motion.—In proportioning the parallel motion of a stationary engine, the radius and parallel rods are generally made of the same length, and this length equal to half the radius of the great beam. The back and main links are generally made equal to half the length of stroke, the air pump cross head being inserted in the back links at the middle of their length, whence an almost perfect parallel motion is obtained for the air pump.

The sectional area of the main links and back links should be made a little more than the area of the piston rod and air pump rod respectively.

Proportions of the Beam.—The depth of the great beam at the centre may be made equal to the diameter of the cylinder, and the depths at the ends equal to two-fifths. The length of the beam should be about three times the length of the stroke. The diameter of the end studs should be one-ninth of the diameter of the cylinder, whether they be formed of cast or wrought-iron. The thickness of iron for the web may be proportioned as follows:—Multiply the pressure of steam on the piston in pounds by half the length of the beam in feet, and divide the product by seventy times the square of the depth at the centre in inches, then the quotient will be the thickness of the web in inches. For small powers, this proportion may be increased a little.

The Connecting Rod, &c.—The length of the connecting rod of a beam engine is usually made equal to three times the length of stroke. When of cast-iron of the usual form, the sectional area at the centre of the rod is made about 1-28th of the area of the cylinder, and at the ends, about 1-35th; and the breadth across the arms of the cross, about 1-20th of the length of the rod. The crank pin may be made 1-7th of the diameter of the cylinder, if of wrought-iron, or 1-6th for cast-iron. The connecting rods of land engines are now frequently made of wrought-iron, as in the marine engine; and, indeed, the side lever marine engine itself is, in many instances, applied as the motive power for mills of various descriptions.

The diameter of the necks or journals of cast-iron crank shafts may be determined as follows:—Multiply the square of the diameter of the cylinder in inches, by the length of stroke in feet, multiply the product by the decimal ·15, and extract the cube root, which will give the diameter of the neck in inches.

The Fly Wheel.—For the weight of the fly wheel, either of the two following rules may be adopted:—Multiply the constant number 1368 by the nominal horse power of the engine, and divide the product by the diameter of the wheel in feet, multiplied by the number of revolutions per minute, the quotient being the weight of the rim in cwts. Or, multiply the horses power by 2000, and divide the product by the square of the velocity of the circumference of the fly wheel in feet per second, the quotient being the weight in cwts.

Tables of the proportions of stationary condensing engines, according to Messrs. Hicks' and Mr. Wm. Fairbairn's practice, respectively, are given in the Appendix, and will be appreciated by the practical engineer.

Speed of the Piston.—The following table, (compiled by Mr. Farey,) agrees with the practice of our most eminent engineers, with regard to the speed of the piston, in proportion to the length of stroke. The effective pressure is more usually assumed at 7 lbs. per square inch of piston.

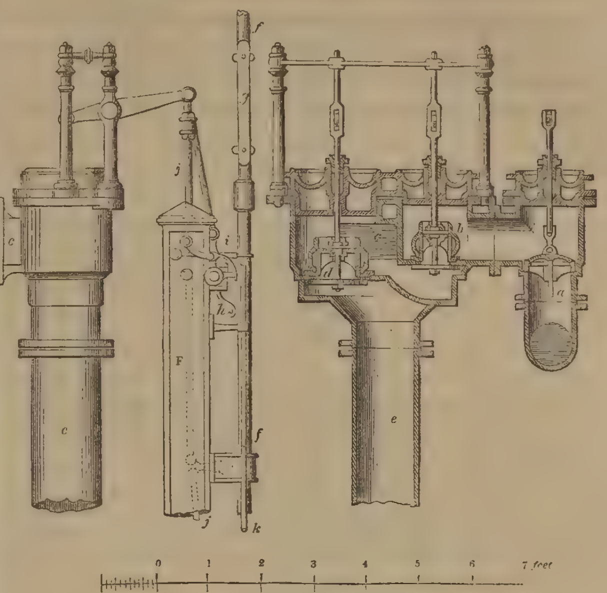
Horse Power.	Diameter of Cylinder.	Length of Stroke.	Strokes per Minute.	Speed per Minute.	Effective Pressure.
No.	Inches.	Feet.	No.	Feet.	Lbs.
4	12	3	29	174	6·8
6	14	3½	27	189	6·82
8	16	4	24	192	6·84
10	17½	4	25	200	6·86
16	21¾	4½	23	207	6 91
20	23¾	5	21½	215	6·92
26	26¾	5½	20	220	6·92
30	28¼	6	19	228	6·92
40	31½	7	17½	245	6·92
50	35⅛	7	17½	245	6·94
70	40¾	8	16	256	6·94
100	48⅝	8	16	256	6·94

II.—*The Cornish or Pumping Engine.*

Gradual Improvement in the Duty of the Pumping Engine.—The mines of Cornwall have been the theatre where the chief improvements in the steam engine have been progressively introduced and perfected; the large quantity of water required to be raised from the mines, and the high price of coals in those districts, having operated as a most powerful stimulus to improvement. Recently, the practice of publishing monthly returns of the "duty" of the engines in Cornwall has been attended with the most beneficial results, for, as Mr. Taylor justly observes, "the proprietors of the mines, whose interests are inseparable from the economical employment of those machines, look to these accounts to govern them in the choice of engines, and in their judgment of those who erect and superintend them." From the time of Savery's and Newcomen's inventions in the beginning of the last century, little was effected for the pumping engine until the era of James Watt, whose improvements, (which commenced in 1776,) soon raised their average duty to 21 millions, or more than double the duty in Smeaton's time. About the year 1780, Watt introduced the system of expansive working, which further increased the duty to 26½ millions of pounds raised one foot high by a bushel, (or about 84 lbs.,) of coals. From this time to the year 1812, little or no advance took place, but between that year and 1814 the duty was raised to 32 millions. In the year last mentioned, Mr. Woolf introduced his double cylinder engine, employing a high pressure of steam and increased expansive action, which advanced the duty to as much as 54 millions. The results obtained from Woolf's double-cylinder engines were in process of time exceeded by Captain Grose's engine at Wheal Hope, which in 1826 had attained a duty of 62 millions. In the following year, however, one of Mr. Woolf's single cylinder engines took the lead by showing a duty of 67 millions. In the year 1827, we are informed that one of Captain Grose's engines, (at Wheal Towan,) reached the high duty of 87 millions, the average duty of 11 engines in Cornwall as reported by Messrs. Lean for that year being 64 millions. Since that time the duty has been further increased by extending the principle of expansion, and using steam at a considerable pressure, so that in 1834 a maximum duty of 90 millions of pounds was attained; and more recently it is asserted to have reached, in some instances 100, 110, and even 120 millions of pounds of water raised one foot high by a bushel of coals. The *average duty* of the *whole* Cornish engines, however, is seldom reported at the present day at more than 55 millions, and the consumption of fuel at 4 pounds per horse power per hour.

Peculiarities of the Cornish Engine.—The Cornish engine differs from the condensing engines we have hitherto considered, in being *single acting;* that is to say, the force of the steam is applied on one side of the piston only, and that during the descending, or as it is called in Cornwall, the *in-door stroke* of the piston, which raises the load. During the ascending or *out-door stroke*, the steam (after it has performed the lift), is admitted from the upper to the lower side of the piston, and, the pressure being equalized, the return stroke is effected by the preponderance of the plungers and pump-rods in the pit. In order to check the velocity at the end of the stroke, due to the accelerated speed of the piston from the descent of the pump rods by gravity, the equilibrium valve closes a short time before the termination of the stroke, and shuts up a portion of steam between the piston and the cylinder cover. This steam is forcibly compressed as the piston ascends, and acts as an elastic cushion to receive and deaden the shock, giving back to the piston at the commencement of the return stroke all the power exerted in compressing it. These operations are regulated by four valves, of which three are indicated at Figs. 22 and 23, representing the upper valves and valve gearing for a Cornish engine constructed at Hayle Foundry. The regulating valve *a*, through which the steam first enters the valve chest, is fixed open to the required degree by a screwed spindle. If we suppose the piston to be at the top of the stroke, the steam valve *b*, is now suddenly opened, and the steam, at 20 or 25 lbs. pressure above the atmosphere, rushes into the upper part of the cylinder through the steam port *c*. The piston accordingly descends, with a rapidly increasing velocity, until it has accomplished a fraction of the stroke varying from one-tenth to one-eighth, one-sixth, one-fourth, &c., according to the degree of expansion at which the engine is desired to work. The steam valve *b*, is then closed, and the piston completes the stroke by the joint effect of its acquired momentum and the expansive action of the enclosed steam.

Fig. 22. Fig. 23.



Nozzles of a Cornish Engine.

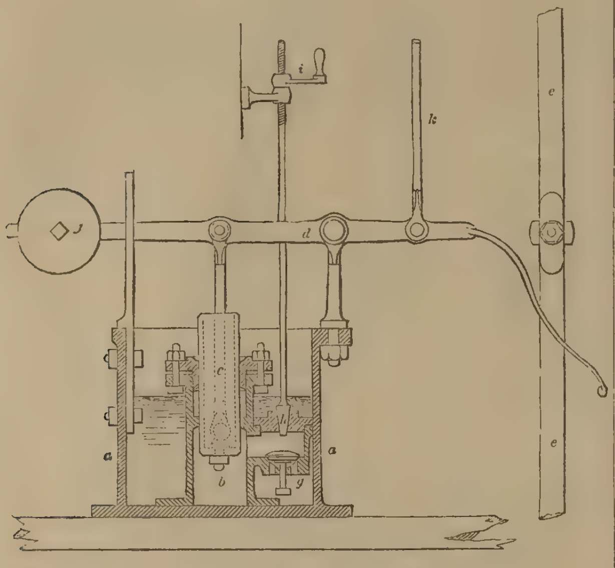
a, the steam pipe with regulating valve. *b*, the steam valve. *c*, steam port to cylinder. *d*, equilibrium valve. *e*, equilibrium or exhaust pipe. E, frame for hand gear. *f*, the plug rod. *g*, tappet on ditto. *h*, lever for opening steam valve. *i*, second ditto. *j*, rod and lever for opening exhaust valve. *k*, rod to cataract.

The Power of the Engine adjusted to the Resistance.—It forms an important feature in the economy of the Cornish engine, that the piston in the descending stroke is brought quietly to rest at the bottom without any shock, in consequence of the power exerted by the steam during the stroke being so regulated as to be exactly equal to the power consumed by the resistance. The degree of expansion to be given in the cylinder is regulated by tappets on the plug rod *f*, which close the steam valve at the proper time. Should more power be generated in the cylinder than suffices to bring the piston quietly to rest at the bottom of the stroke, a cross bar, attached to the cylinder end of the great working beam of the engine, strikes forcibly against two wooden blocks fixed on the spring beams, which thus receive the shock of the piston, and prevent its striking the cylinder bottom. In such a case, either the regulator valve is closed a little, so as to throttle the steam, or the degree of expansion is increased, until the end of the working beam strikes lightly on the spring beams. On the other hand, if the pressure on the piston be too much diminished, it completes the stroke prematurely, and the lift of the water in the pumps is proportionably diminished.

The Speed regulated by the Cataract.—During the whole period of the descending stroke, the injection water has been flowing into the condenser, as in the ordinary condensing engine; but before the commencement of the return stroke the flow of injection must be stopped, until the exhaustion of the steam again commences with the descending stroke. The opening and shutting of the injection valve must therefore be regulated by appropriate mechanism.

The contrivance universally adopted for this purpose, as well as for opening the steam and exhaustion valves of the single-acting engine, is the "cataract," which thus also determines the interval which is allowed to elapse between the successive strokes of the engine, in order that the number of strokes per minute may correspond with the quantity of water to be drawn from the mine. It mainly consists of a pump-barrel and plunger, being in fact a little plunger forcing pump. The inlet to the water is through a valve opening freely upwards, but the outlet is contracted at pleasure by means of a movable plug. The plunger is attached by a joint to the centre of one arm of a horizontal lever, weighted at the extremity, and communicating through the other arm with the steam valve and the exhaustion and injection valves. Now, when the cylinder piston has descended nearly to the bottom of its stroke, a tappet on the plug rod, at the bottom, strikes one end of the horizontal lever, and thus raises the cataract plunger, the water at the same time freely entering the barrel below the plunger, from a cistern in which the whole apparatus is placed. When the stroke is finished, and the piston and plug rod begin to ascend by the preponderance of the weight in the pit, the tappet, which has just raised the plunger, quits that end of the horizontal lever, and the weight fixed at the end of the opposite arm now comes into play, and expels the water from the pump, by forcing the plunger down. But the inlet valve having closed, the only exit for the water is by the aperture left round the conical regulating plug, so that by augmenting or diminishing the area of this aperture, the exit of the water, and thereby the descent of the plunger, may be accelerated or retarded at pleasure.

Fig. 24.



Cataract for a Cornish Engine.

a, the cistern. b, the barrel of pump. c, plunger of ditto. d, lever for working plunger. e, plug rod of engine. f, weight for depressing the plunger. g, valve for admission of water into the barrel. h, regulating conical plug. i, screw for adjusting the regulating plug. k, the rod communicating with the hand gear of the engine.

The regulating plug of the cataract is therefore made capable of being adjusted, to the utmost degree of nicety, by a micrometer screw and handle, as upon the area of aperture which it leaves for the exit of the water from the barrel depends the length of interval between the strokes of the engine, and therefore the number of strokes per minute. For, on the instant when the cataract plunger has reached the bottom of its stroke, by forcing all the water which opposed its descent through the regulated aperture, a rod connected with the cataract lever liberates a weight, which in its fall opens, at the same moment, the exhaustion and injection valves. The steam now rushes from the cylinder into the condenser, a vacuum having been formed under the piston a little before, and preparatory to, the admission of the steam above it. The same rod which opened the exhaust and injection valves, continuing to ascend, now suddenly opens the steam valve in the same manner as the other two, namely, by releasing a weight on the removal of a catch. The suddenness of the action is considered a great advantage, especially with regard to the *admission* of the steam, which then exerts what has been called a "percussive action" in starting the piston on its way. The stroke then proceeds as we have already described; the steam valve *b* (Figs. 22, 23), being first closed; then the exhaust and injection valves together; after which the equilibrium valve *d*, is opened so as to permit the steam to pass round by the pipe *e*, to the other side of the piston, which is thus brought into a state of equilibrium; and finally, the cataract plunger *c*, Fig. 24, is raised preparatory to the next descending stroke.

Causes of the Economy of the Cornish Engine.—The principal causes which operate towards the wonderful economy of steam exhibited by the Cornish Pumping Engine, may be thus enumerated:—1st, the practice of employing high-pressure steam in addition to the power obtained by condensation; 2ndly, the great extent to which the principle of expansion in the cylinder is carried, and the additional advantage derived from expanding high-pressure steam; 3rdly, the perfect manner in which the whole power developed within the cylinder is transmitted from the piston to the working beam, without any of the momentum acquired during the stroke being destroyed upon the fixed parts of the machine; 4thly, the comparative simplicity and fewness of the working parts of the engine, so that the loss of effect from friction is small; 5thly, the interval of rest which is allowed to elapse between each stroke, in conjunction with the large area of the eduction or exhausting passages, by which the vacuum is rendered more effective; 6thly, the system which prevails of stopping the radiation of heat by every possible means, namely, by surrounding the cylinder and cylinder cover, &c., with a steam jacket, itself protected by non-conductors from the external air, and by the perfect clothing of the boilers and steam pipes; 7thly, the superior form and proportions of the boiler and furnace for extracting the heat from the products of combustion. And lastly, the careful management of the fires, and the thorough combustion of the carbon of the fuel by not overloading the bars.

Peculiarities of Construction.—We shall now describe more minutely some of the peculiarities of construction of the Cornish engine. The cylinder is surrounded by a cast-iron casing or jacket, which leaves a space of about one inch between itself and the outside of the cylinder. This space, which communicates with the boilers by a small pipe about four inches diameter, is kept constantly full of high-pressure steam, and thus preserves the cylinder at a high temperature. This is considered peculiarly advantageous in working expansively, as the steam in the act of expanding, parts with its own sensible heat, and becomes very liable to condense upon the sides of a cool cylinder. Besides, any particles of water which the steam may suspend will thus be converted into vapour by the high temperature, as the pressure diminishes during the expansion of the steam in the lower portion of the stroke. As the boilers are usually placed at a lower level than the bottom of the cylinder, any water which may condense within the jacket is conveyed back to the boiler (in a hot state) by a small pipe.

Several attempts have been made to discontinue the use of the steam jacket, but they have been always attended with a manifest loss of effect within the cylinder. The steam jacket itself is surrounded by a non-conducting envelope of ashes or sawdust, bricks and mortar, wood, &c., which so effectually prevents the conduction of heat, that in the best Cornish engines the temperature of the outside casing seldom exceeds 70° or 80°; and the temperature of the engine house is raised but a few degrees above that of the external air. The cylinder cover is made hollow, and used formerly to be kept full of steam, in the same manner as the cylinder jacket; but this practice having been found very inconvenient, from the necessity of breaking the joints of the steam pipes each time the cover was raised, the cavity is now usually filled with non-conducting materials, as hair, sawdust, &c., which effectually answers the purpose of preventing radiation. The steam is admitted by a small pipe into a vacant space below the bottom of the cylinder. The piston rod stuffing-box is also formed in a peculiar manner. It is made very deep, and about the centre of its depth there is formed, between the upper and lower packings, a small hollow chamber, by means of two brass rings encircling the rod, and held apart by little pillars, or distance pieces. Into this chamber steam is admitted through a small pipe, communicating with the cylinder jacket, the object being to prevent the leakage of air into the cylinder through the stuffing-box, when the pressure of steam within falls below the atmospheric pressure, as would be the case during the greater part of the stroke while working expansively.

Speed of the Pumps.—It is found advantageous in the Cornish engine not to allow the pumps the full speed of the piston. The centre of the working beam is accordingly placed a little nearer the pump end, the usual proportion between the stroke or velocity of the piston and the pump plungers being as 10:8.

Cornish Double-beat Valves.—The description of steam valve universally employed in Cornwall is that known either as the equilibrium valve; or, from its two conical ground faces, the double-beat valve; or it is sometimes called, *par excellence*, the Cornish valve. It is now generally employed in all cases where the difference of steam pressure above and below the valve is so great as to present a serious difficulty in opening a valve of the common construction.

This valve is represented in section, at wood-cuts, Figs. 22, 23, as applied to a Cornish engine; at Plate CXXXIII., as applied to a stationary engine, by Mr. Fairbairn; and at Plates LXXIV., LXXVII*., as applied for the expansion valve of a marine engine.

By examining the valve it will be seen that, when on its seat, the only pressure opposing its upward motion is that due to the horizontal areas of the two ground faces, which is comparatively of small amount, the pressure due to the rest of the horizontal area of the steam passage being borne by the fixed plate of the valve seat. Thus the double-beat valve requires but a small power to open it, and the gearing for this purpose may be made light and simple; whereas, to open a common pot-lid valve of, we will suppose, 12 in. diameter, against a pressure of 15 lbs. to the square inch, requires a force equal to three-quarters of a ton, with gearing to correspond. The valve itself is simply a solid ring of gun-metal, on which are formed the two conical beating surfaces which exclude the steam, the valve spindle being fixed in a central boss, supported by radial arms or feathers cast on the upper part of the valve. The valve seat is a hollow cylinder, close at the top, and open at bottom, with the sides perforated so as to allow the steam, when the valve rises on its seat, to flow through the interstices, and to take the course indicated by the arrows in Plate CXXXIII. The upper steam nozzle is protected from the cooling influence of the external air by a metal casing, which is filled with sawdust, ashes or other non-conducting material.

The eduction pipe in a Cornish engine has usually an area equal to twice that of the steam pipe. The condenser is placed in the open air, outside the engine house. The air pump has usually half the stroke of the cylinder, and is worked from the pump end of the beam; so that its working stroke is made at the same time with that of the cylinder, extracting the water and vapour from the condenser, as the piston descends, and the vacuum space becomes lessened.

Pit Work.—In arranging the work of the pumps, the whole depth of the mine is divided into separate lifts of about 30 to 40 fathoms each, each stage in the pit being provided with its own pump and cistern, into which the ascending pipe of the next lift dips. The pumps employed are invariably plungers for all the lifts except the lowest. Hence, the greater part of the water is forced up the ascending pipes by the gravity of the pump rods, the power of the engine being exerted in raising them again during the descending stroke of the piston. The main pump rod is formed of stout baulks of Memel timber, bolted together endways, and the plungers are worked at the different stages by side rods strapped on to the main one. Plunger pumps are preferred in the mines to buckets, as they require much less repair, and they are also more convenient from the fact of their performing their work during the *descent* of the pump rods, so that a much less counterbalance is required than if all the water had to be lifted by the common pump bucket. We have said that the bucket is employed for the first or lowest lift, and for this reason, that, should the water from any accident rise above its usual level in the bottom of the mine and cover the pumps, a plunger could not be got at for examination or repair if out of order, whereas a bucket can be drawn up the ascending pipe to the first stage or level, and replaced without difficulty when required. It is also more convenient than a plunger, when it is required to lengthen the pipes, in order to meet an increase of depth in the mine.

An "adit," or tunnel, is usually cut from the side of the pit to a neighbouring valley or low level, through which the drainage water of the mine may flow off without the necessity of raising it to the surface of the ground. Care is also taken to conduct the drainage from the sides of the pit into the nearest pump cistern, so as to avoid lifting it from the bottom of the pit, or any higher than necessary. On this account, the ascending pipes near the top are made a little larger than at the bottom of the pit. It frequently happens, that the gravity of the pump rods exceeds the force required for working the plungers, in which case, a counterweight is employed to balance the surplus weight of the rods. The counterbalance is applied by loading one extremity of a lever or scale beam turning on a fulcrum, the other extremity being attached to the main pump rod.

Thus, in the case of Davey's engine at the Consolidated Mines, the main rod is one-third of a mile long, and weighs 95 tons. The other rods weigh 40 tons, total 135 tons. Of this immense weight, which is an extreme case, only 39 tons are required for working the pumps, the remainder being balanced partly by counterweights, and partly by hydrostatic pressure.

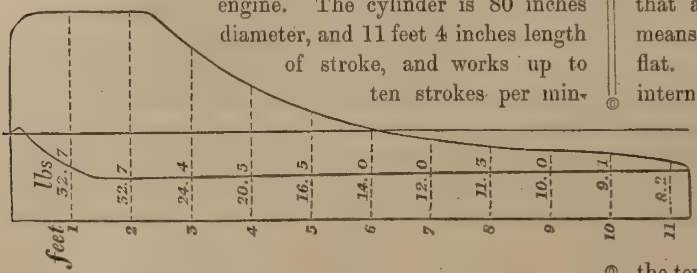


Fig. 25 exhibits an indicator diagram taken from this engine. The cylinder is 80 inches diameter, and 11 feet 4 inches length of stroke, and works up to ten strokes per minute, when necessary. The load on the piston is 13·12 lbs. per square inch, and the average duty about 70 millions. The temperature of the injection water is 83½°, owing to central heat from the depth of the mine; and of the hot well 110°; the barometer in the condenser showing a vacuum of 27¾ inches. It would appear from the diagram that the steam valve is not closed until the piston has made about 27 inches of the stroke, or one-fifth of the whole, after which the expansive action commences. The mean pressure on the piston, during the stroke, will be found to be about 17·4 lbs. per square inch, thus distributed:—

Distance in feet from top of Stroke.	Pressure in lbs.
1	32·7
2	32·7
3	24·4
4	20·5
5	16·5
6	14·0
7	12·0
8	11·3
9	10·0
10	9·1
11·33	8·2

Length of Stroke, 11 ft. 4 in.
Mean Pressure, 17·4 lbs.

Speed of the Engine.—The best Cornish engineers consider the most advantageous velocity for the in-door stroke of the engine to be about 250 feet per minute, and for the out-door stroke about 140 feet. The number of strokes per minute depends, as we have said, upon the weight of water to be raised, but the maximum effect is believed to be obtained by working the engines at 9 or 10 strokes per minute.

CHAP. IV.—LAND ENGINE BOILERS.

Having previously considered at some length the principles and construction of boilers for marine purposes, it will be apparent that much of what has been already said on this subject applies with equal force to boilers of every description. It only remains, therefore, to advert briefly to the peculiarities in the construction and management of land engine boilers which their varied circumstances require.

Approved Forms.—At present, the prevailing and most approved form is either the waggon-shaped boiler introduced long since by Boulton and Watt, and represented at Plates XXIII., XXIV., or the horizontal cylindrical boiler, made either without or with an internal circular flue, as employed in Cornwall. It is not without good reason that these varieties have come so generally into use both for high and low pressure steam, although for high steam the cylindrical boiler is evidently the most appropriate. In both cases, the heating surface is capable of being made highly effective by the easy escape of the steam from the internal surface of the plate; the forms are simple and economical in point of material, and the convex surfaces impart strength without the necessity for *stays*, which should be always avoided when possible. It must be borne in mind, however, that a surface convex *towards the steam pressure* is by no means a strong one, and requires support if the curve be flat. A cylindrical boiler with hemispherical ends and an internal circular flue of 20 to 30 inches diameter, is manifestly independent of stays in any position.

Extent of Flue Surface.—Although, in a theoretical point of view, it appears desirable to extend the flues of a boiler until the temperature of the contained air, before entering the chimney, shall have been reduced to the temperature of the steam in the boiler, it is found in practice that, in ordinary cases, the increase of evaporative power thus obtained is not commensurate with the extra cost of material and labour in the boiler and setting. Thus, it has been proved by direct experiment that the loss of fuel resulting from even a considerable diminution of heating surface, caused by cutting off 10, 20 or even 30 feet from the length of a long cylindrical boiler, has amounted to a very small per centage indeed upon the first cost of the extra boiler plate and brick work. This remark chiefly applies to cases where fuel is not very expensive, as in the manufacturing districts, and would require modification if applied to the case of boilers in Cornwall, where any saving of fuel, however effected, is perhaps to be preferred to every other consideration. It is found, however, in Lancashire that an effective heating surface of 10 to 12 square feet per horse power is as much as can be economically allowed.

Grate Surface.—The average amount of grate-bar surface is about one square foot per horse, this proportion varying considerably according to the views of the maker on the comparative advantages of slow or rapid combustion. We believe, however, that the proportion just named will be found an advantageous one. If the grate surface be made very small, the fires require constant stoking, by which means the coal is prevented from caking properly, and falls in considerable quantity through the interstices of the bars, this loss being increased by the necessity of keeping the bars further apart than would otherwise be desirable, in order to admit the requisite amount of air for supporting combustion. Also, the most efficient heating surface in the boilers,—namely, that immediately over the fire,—is diminished, and a good deal of carbon escapes in smoke during the frequent opening of the door for stoking. We may observe that it is frequently advantageous to leave the fire door *a little* open, in order to admit a portion of pure air to complete the combustion of the gases in the furnace; but if too much cold air is thus allowed to enter, it cools down the gases so much as to *prevent* their due combustion, besides the injurious effect of cooling the contents of the flues. If, on the other hand, the grate surface be made very large in proportion to the flue surface, the danger is either that a large portion of the heat will escape up the chimney before it has been extracted by the flues, and transmitted to the water of the boiler, and that, should the fuel be kept thin on the bars (which is the usual practice in such a case), an inordinate quantity of air will enter and cool the flues; or else, in order to check the rapidity of the combustion, the draft must be diminished, and oxygen be supplied more sparingly to the fuel; the result of which is that, the combustion being now languid and stifled, smoke is formed, and a portion of the combustible gases escapes up the chimney unignited, or perhaps takes fire and burns at the chimney top. It will be seen from these considerations how much of the economy of the boiler depends upon the care, the intelligence, and the experience of the fireman, as to the proper regulation of the draft, keeping the interstices of the bars open and free from dust and clinkers, watching over the caking and subsequent combustion of the fuel, and returning to the fire the cinders which may have fallen through between the bars, &c.

Form of the Furnace.—It is found that *space* in the furnace is necessary for the proper commingling of the inflammable gases with the atmospheric air, and also that heating surface in the immediate vicinity of the fire is very valuable for raising steam; hence it follows that the furnace should be so large, and of such a shape, as to facilitate the mingling of the gases in the first instance, and to present a large surface to the flame after they are ignited. The length to which the flame extends through the flue of a boiler (or along the bottom, as the case may be), depends very much upon the intensity of the draft and the management of the fires. Thus with a sluggish draft, or where the bars are choked with clinkers, or over-loaded with fuel, a dull red smoky flame, like that from an unsnuffed candle, is produced, which extends a long way through the flue with very little heating power; whereas with a smart draft and clean bars, a bright flame of great intensity is the result, which extends a shorter distance, but gives out much more heat than the preceding. The distance to which the flame extends in a well-constructed boiler is seldom more than three times the length of the fire bars. Where the draft is deficient either from mal-construction of the boiler or setting, or from want of power in the chimney, a fan blast has sometimes been resorted to with advantage, but this is by no means to be recommended under ordinary circumstances, as the fanners absorb a large quantity of power from the engine.

Smoke Burning.—What we have said on the subject of smoke burning, in connection with marine engines, applies equally well here; the point to be aimed at being, not to consume the smoke, but to prevent its formation in the furnace by supplying a quick draft, by allowing the boiler plenty of grate surface and space over the bars, and by ensuring a skilful management of the fires.

We are confident that a proper degree of attention bestowed on these points in the original construction of the boiler, and in the skilful management of the draft and firing, will do more to prevent smoke, and economize fuel, than all the so-called "smoke burning" apparatus that ever was, or will be, invented.

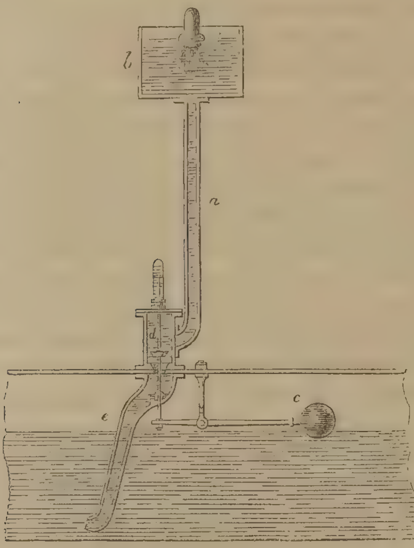
Steam Capacity and Priming.—The capacity for steam in the boiler is recommended to be not less than 12 or 14 cubic feet for each horse power. This space is necessary to prevent "priming" into the cylinders, but should the water be very dirty, and therefore liable to form a slimy deposit within the steam chest, the capacity may still be augmented with advantage. What is termed *priming*, or the boiling over of the water into the steam pipe, is chiefly occasioned by the sudden abstraction of the steam at the commencement of each stroke of the piston, and the consequent violent ebullition of the water from the pressure within the boiler being suddenly diminished; hence, any increase to the steam room acts beneficially by calming these dangerous oscillations of pressure, to which are to be attributed not only priming, but in some instances, the explosion of the boiler itself. Dirty water, such as that usually obtained in large towns, increases this action very much; so that it is necessary to clean out the boiler at least once a month, and in many cases, once a fortnight, or even weekly, to prevent an undue accumulation of slime, which adheres to the steam chest at and above the water level. Mechanical means for cleansing the boiler of this filth, without the necessity of disusing and emptying it out, (which is the usual practice,) have been sometimes adopted with success. It is advisable, however, to have one spare boiler more than is necessary for supplying steam to the engines, so that each boiler may be alternately disused in rotation, for cleaning, examining and repairing, without any detriment to the supply of the engines. A little soap thrown into the boiler is one of the best *alleviators* of priming, but there is no *preventive* except clean water and plenty of steam room.

Water Space.—The amount of water contained in the boiler is not of much importance, except in so far as it determines, in a measure, the quantity of fuel consumed in getting up the steam before starting the engine; but the area at the water level should be made as large as convenient, so that the surface may not be too much disturbed by boiling, or subjected to much alteration of level by a temporary neglect or stoppage of feed.

Chimneys.—The subject of draft in chimneys is one which presents much difficulty to a theoretical investigation, but we are enabled from practice to determine, with tolerable certainty, what are the most advantageous proportions for the chimney of a stationary engine. It is found that little or no advantage is gained in point of draft by carrying the chimney to a height of more than twenty-five or thirty yards, although it is frequently desirable to do so in order to spread the smoke and deleterious fumes. For chimneys of this height, in Lancashire, it is a usual and a good rule to allow an area at the top of twenty square inches per horse power; but if the chimney is higher than this, a larger area should be allowed. The draft should have sufficient power to raise about half an inch of water in a small glass syphon or pressure gauge, inserted at the bottom of the chimney, one leg within, and the other exposed to the external air.

Feeding Apparatus. — Fig. 26 represents the usual method of feeding a boiler generating low pressure steam, the water in the upright pipe *a*, and the cistern *b* (or *Jack-head*, as it is technically called), descending into the boiler against the pressure of steam by its own gravity, the *quantity* being regulated by a float-stone *c*, and valve *d*, which admits a supply of feed according as the water level sinks in the boiler; the pipe *e*, being employed to convey the feed water to that part of the boiler where there will be least agitation. The same general arrangement is shown in another form in Plates XXIII., XXIV.

Fig. 26.



A high pressure boiler, on the other hand, requires to be fed by means of force pumps worked by the engine, which inject into it a portion of water at each stroke, the surplus, which is rejected by the boiler, passing away through an escape valve. Such an arrangement is represented by Figs. 27 and 28, which are two of the most approved methods for feeding high pressure boilers.

Fig. 27.

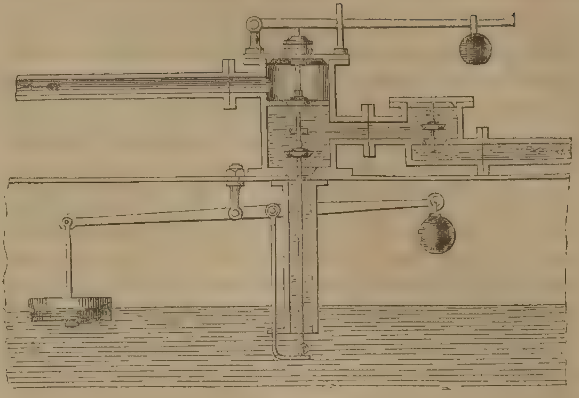
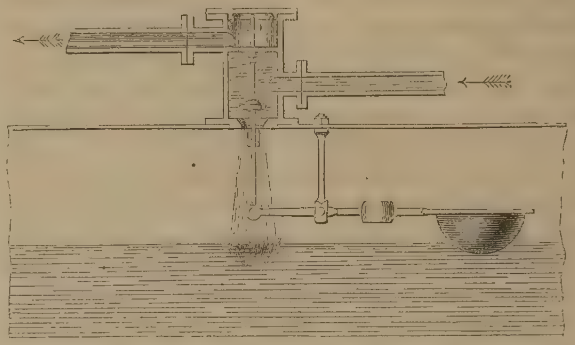
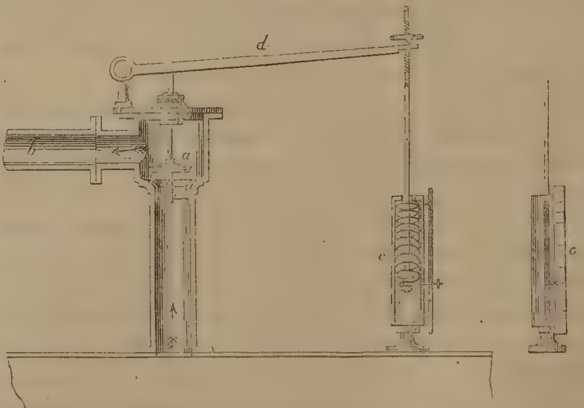


Fig. 28.



Safety Valves.—Every boiler should be fitted with two safety valves of sufficient area, one under the control of the engine tenter, and the other placed beyond his interference. In the case of low pressure boilers they are loaded with weights, but as these would be inconvenient with steam of high tension, a spring balance is then usually applied to the extremity of the safety valve lever, as in Fig. 29, where *a* represents the safety valve; *b*, the waste steam pipe; and *c*, the spring balance, graduated in the usual manner to show the pressure, and acting at the extremity of the lever *d*, against which the safety valve spindle abuts. This arrangement, which is invariably applied in locomotive engines, is also represented in Plates LXVI.—LXXIII.

Fig. 29.



Evaporative Power of Cornish Boilers.—The following experiment, on the evaporating power of a Cornish boiler, was lately made at Par Consols mine, by direction of Sir Henry De La Beche and Dr. Lyon Playfair, for the Parliamentary report on coals, in order to compare with the results obtained from their experimental boiler, as already described. The boilers on which this experiment was made, are each 32 feet long, and 6 feet 3 inches diameter. Each boiler presents a heating surface of 950 square feet, and is provided with an arrangement by which the feed water is heated to near the boiling point, before entering the boiler. This is effected by means of the waste heat escaping from the flues, the apparatus consisting of two wrought-iron tubes, each 20 inches diameter, with a surface of 560 square feet, placed one above the other, parallel to the length of the boilers. The feed water is pumped by the usual arrangement into the upper tube, and then descends through a pipe into the lower one, from whence it passes into the boiler. Both tubes are exposed in their whole length to the action of the heated gases, after they have completed the circuit of the boiler; their temperature at the base of the chimney being thus reduced to about 300°. The engine at Par Consols mine has an 80 inch cylinder, and 12 feet stroke, and is worked by two boilers, as described. After a lapse of 46½ hours, it was found that 11,730 lbs. of coal had been consumed in evaporating 119,700 lbs. of water, from 92° Fahr.; which gives 10·204 lbs. of water evaporated from that temperature by one pound of coal. If we take 212° as the standard temperature, the weight of water evaporated by one pound of coal will be 11·428 lbs. The coal employed was a mixture of Swansea and Bury coal, containing 84 per cent. of carbon, 4·2 of hydrogen and 8 of ash.

The trial with the experimental boiler which was considered to afford the fairest comparison with this trial, gave a result of 9·52 lbs. of water, evaporated by one pound of coal, thus proving the evaporative values of the two boilers to be as 9·52 : 11·42; or a superiority of nearly 20 per cent. in favour of the Cornish boiler.

CHAP. V.—THE LOCOMOTIVE ENGINE.

This variety of the steam engine, which has of late years advanced to such a wonderful degree of perfection, employs the agency of steam in the simplest form in which it is capable of producing power—namely, by the mere excess of its pressure above that of the atmosphere. The reason why the more economical process of condensation is inapplicable in this case is obviously the increase of weight and complexity attending the use of that process, both of which should be avoided as much as possible in a machine which propels its own weight, at an immense velocity, over artificial works, composed of yielding materials, all more or less liable to wear and tear from the effects of momentum and friction. The qualifications for a perfect locomotive are not easily reconcileable with each other; and demand the utmost skill, combined with long experience, on the part of the manufacturer.

We have said that lightness and simplicity of parts are desirable; but these, on the other hand, must not interfere with the safety and the economical working of the engine—its stability must be ensured by the position of the wheels, and of the centre of gravity—and every available means must be adopted for increasing, first, the evaporative power of the boiler, and secondly, the mechanical effect of the steam, without unduly increasing the weight and cost of the apparatus. It is evident that any increase in the economical working of the locomotive, effects a twofold saving—namely, in the cost of the fuel, and in the weight to be carried. Hence, the steam is worked at a very high pressure, (from 80 to 100 lbs. above the atmosphere), and is cut off and expanded in the cylinder in the most economical manner; as will be afterwards explained.

As a detailed description (with Plates) of Messrs. Hawthorn's locomotive engine will be found in the body of this work, the reader is referred to it for a knowledge of the form and disposition of the various parts; and we shall now proceed to notice some of the principal dimensions and leading features of locomotive engines in general.

Locomotive Boiler.—The boiler, which here, as elsewhere, is the very soul of the engine, first claims our attention. This is divided into a cylindrical shell, or body, with internal flue tubes; having attached to it at one end the furnace, or internal fire-box, and the external fire-box; and at the other end the smoke-box and chimney.

It is fitted with a small steam chest, or *dome*, feed pipe and cock, two safety valves, steam gauge, glass water gauge, three water gauge cocks, steam whistle, man hole and mud holes, &c. The diameter of the cylindrical body is usually made from 3 feet 3 inches to 4 feet on the narrow gauge, and even 5 feet on the broad gauge, and from 9 to 12 feet long, according to the diameter of the cylinders, and the power of the engine. The thickness of plate is generally $\frac{5}{16}$ inch, sometimes $\frac{3}{8}$ inch; the flat ends being stayed by longitudinal rods.

Tubes.—The tubes are usually formed of brass, from 1½ to 2 inches external diameter, with three-fourth inch water spaces between each; a considerable space being left below the bottom row of tubes for deposit.

Brass tubes, although much more durable than the old material of copper, are still very rapidly worn out by the friction of the ashes passing through them; and iron tubes have lately been successfully introduced by some manufacturers. They are much less expensive, and, where the water is tolerably pure, last longer than the brass ones. Their heating surface, however, is less effective in the ratio of the conducting powers of the two metals. Brass tubes cannot be securely fastened without the use of internal steel rings, or ferrules, which act injuriously by contracting the area for draught through the tubes; whereas, wrought-iron tubes may be secured by hammering the ends (by means of a conical die) into a countersunk recess formed in the tube-plate.

The practice of giving an increased length to the boiler of the locomotive is now becoming general on the narrow gauge lines, affording, as it does, an increase of power as well as of economy to the machine. A great length of tube is especially valuable at high velocities, when the intensity of the fire is so much augmented by the rapid respirations of the blast pipe. We do not agree, however, with the opinion of those who allege that the effect of a quick draught is to draw out the flame to a greater length, and on that account, that the long flue tube becomes necessary to absorb the radiated heat of the flame. We believe that the effect is rather to concentrate and confine the combustion of the inflammable gases to the fire-box and the entrance of the tubes, where the flame is the more intense as it is short-lived, (as exemplified in the common argand burner;) the additional length of tube being required to absorb, by contact, the heat from the uninflammable gaseous products, which may be supposed to have acquired a temperature proportional to the intensity of the flame at the mouth of the tubes. Although economy of fuel is thus promoted, it is well known from experience that a large area of grate bar, and extensive heating surface in the fire-box, are the surest means of augmenting the evaporative power of the boiler. Hence, the best practice appears to indicate the propriety of employing a considerable grate bar surface, a roomy fire-box, with extended heating surface, and moderately long tubes.

Fire-Box.—The fire-box, which is in contact with water at the top, sides and back, is usually formed of copper—this material being less liable than iron to be injured during the process of bending the plates and hammering the flanges for connecting it with the rest of the boiler. Also, from its possessing a superior conducting power; copper transmitting the heat of the flame to the water more rapidly than iron. Of late years iron fire-boxes have, in some instances, been introduced, and are said to have proved as effective, and more durable and economical, than the copper ones—the utmost care being of course necessary to select the very best material, and secure the most skilful workmanship. The fire-box is usually made square, and of considerable depth, in order to obtain as large an extent of heating surface as possible—the value of an additional foot of heating surface in the furnace being usually considered equal to three feet in the tubes. The requisite strength is given to the square fire-box by stays, four or five inches apart. Circular fire boxes are also in use, especially on the broad gauge lines, where the manufacturer is not so confined for boiler room. They still require stays, however, to guard against the rapid and unequal waste to which they are exposed by the intense heat, which will warp the plates and destroy their original spherical form. The tube plates are usually of copper, sometimes of iron, $\frac{3}{4}$ to $\frac{7}{8}$ inch thick. The bursting of a worn or defective tube is the most common accident to which a locomotive boiler is subject. A number of wood or iron plugs are therefore always carried, one of which is driven into each end of the defective tube as soon as the water of the boiler has run out to the level of the fracture.

Fire Bars.—The fire bars in the locomotive boiler should be made of wrought iron, very thin and deep, and placed close together. They will thus best withstand the intense action of the fire; and the small interstices between them will prevent an undue waste of coke from the cinders falling through. It frequently happens that the fire has to be suddenly withdrawn from under the boiler, to prevent burning the plates, in case of the water being blown out. This is effected by introducing a dart-shaped poker between two of the bars, and lifting them out of their positions, when the fire is raked through the opening thus made into the ash box.

Proportions of Locomotives.—The external fire-box is formed of iron plate, $\frac{5}{16}$ to $\frac{3}{8}$ inch thick; the smoke box, $\frac{1}{4}$ inch; and the chimney, $\frac{1}{8}$ inch. The latter has an area of about $\frac{1}{10}$ of the grate bar surface, and is 14 feet above the rails. A common rule is to give it a diameter equal to that of one of the cylinders.

The steam dome should be made from 24 to 30 inches high, and about 22 inches diameter. The steam room in a locomotive should be equal to the cubic capacity for water, or nearly so. The diameter of each safety valve is usually found by dividing the diameter of the cylinder by 4. The diameter of each feed pump plunger may be obtained by dividing the square of the diameter of the cylinder by 100. The stroke of the ram or plunger, being taken from the piston cross-head, is equal to the stroke of the cylinder, which has been made, by common consent, 18 inches for most locomotives. The diameter of the cylinder varies from 12 to 18 inches, 16 inches being at present the maximum on the narrow gauge, and 18 inches on the broad.

The square of the diameter of the cylinder in inches multiplied by ·03, will give the usual diameter allowed for the main steam pipe. The area of each steam port in square inches may be found by multiplying the square of the diameter of the cylinder by ·07, and of the eduction port by multiplying by ·125. The diameter of the piston rod should be made $\frac{1}{7}$ of the diameter of the cylinder; and that of the connecting rod in the middle $\frac{1}{5}$.

Subjoined will be found the principal dimensions and weight of some of the most powerful engines now in use.

PASSENGER ENGINE, GREAT WESTERN, (BROAD GAUGE).

Area of fire grate	21 sq. feet.
Area of fire-box	153 —
Area of tubes	1800 —
Number of tubes	305

Cylinder, 18 inches diameter by 24 inches stroke.
Diameter of driving wheels, 8 feet; of small wheels, 4 feet 6 inches.

Weight of the boiler, complete	12 tons	0 cwt.
— cylinders	1 —	10 —
— outside framing	2 —	10 —
— axles	1 —	17 —
— wheels	6 —	0 —
— sundry other parts	7 —	4 —
Total weight	31 tons	1 cwt.

Of this weight, the trailing wheels support	7 tons	3 cwt.
— the driving wheels	10 —	18 —
— four leading wheels	13 —	0 —
Total weight	31 tons	1 cwt.

GOODS ENGINE, GREAT WESTERN, (BROAD GAUGE).

Area of fire grate	18·5 sq. feet.
Area of fire-box	123·8 —
Area of tubes	1231 —
Number of tubes, 2 inches diameter,	218

Cylinder, 16 inches diameter by 24 inches stroke.
Six wheels all coupled, five feet in diameter.

Weight of the boiler, complete, about	10 tons	0 cwt.
— cylinders, complete,	1 —	0 —
— outside framing	1 —	6 —
— axles	1 —	12 —
— six wheels, 5 feet diameter	5 —	2 —
— sundry other parts	5 —	13 —
Total weight	24 tons	13 cwt.

Of this weight, the trailing wheels support	8 tons	2 cwt.
— the centre wheels	8 —	8 —
— the leading wheels	8 —	3 —
Total weight.	24 tons	13 cwt

PASSENGER ENGINES, NORTH WESTERN, (NARROW GAUGE).
Cylinders, 16 to 18 inches diameter and 21 inches stroke.
Wheels, from 6 feet to 8 feet diameter.
Total weight from................................22 to 25 tons.

Weight on driving wheels, from	12 to 14 —
— on leading wheels	6 to 7 —
— on trailing wheels	4 to 5 —
	22 to 25

GOODS ENGINE, NORTH WESTERN, (NARROW GAUGE).
Cylinders, from 16 to 18 inches diameter by 24 inch stroke.
Wheels, from 4 feet 6 inches to 5 feet diameter.
Total weight of the engine, from.................24 to 30 tons.
Greatest weight on one pair of wheels..........10 to 12 —

Railway Wheels.—The manufacture of the wheels for locomotives is one requiring the greatest care and skill in the execution. The usual mode of construction is to forge the spokes or arms of wrought-iron, each with a cross piece or T head at one end, these pieces being afterwards welded together, end to end, to form the rim. In some instances the rim is formed by riveting in place of welding. The centre or nave of the wheel is of cast-iron, which is run round and incloses the inner ends of the spokes. The periphery of the wheel is now turned to the proper size preparatory to being fitted with the tire, which is formed usually of wrought-iron, but sometimes of steel; the best practice, perhaps, being that introduced by Mr. Gooch on the Great Western Railway, where the rim is formed of wrought-iron, about 1¾ inches thick, out of which a dove-tailed groove is cut to receive a band of steel, secured by hammering the metal into the corners of the dove-tail. The tire having been properly attached to the wheel, in the first place by *shrinking on*, and then by countersunk rivets, is then turned slightly conical, in order to facilitate the passage of the engine round curves, the rails being set with a corresponding inclination *inwards* towards the centre of the line. In Mr. Stephenson's six-wheeled engines, the two middle or driving wheels have the periphery flat, without flanges, to diminish the friction in passing curves; all other railway wheels having flanges on the inner edge, to keep the carriages on the rails. The flanges are turned with the corners well rounded, so as to prevent lateral friction on the rails as much as possible. The tires occasionally give much annoyance by separating from the rim, especially when rendered brittle by frost, and as the pieces usually fly off with an immense velocity, some serious accidents have been the result.

The size of the driving wheels must be proportioned to the load proposed to be drawn, as well as to the diameter of the cylinder and the power of the boiler. The wheels cannot be made too large for the economical working of the engines, provided proper proportions be maintained throughout, and the centre of gravity of the machinery be not unduly raised. It has been frequently proved that large driving wheels, combined with a small boiler, are useless; for the first consideration in designing an engine is to calculate the amount of traction required at the rim of the wheel, the diameter being then regulated according to the power of the boiler to produce this traction. The sizes of driving wheels now in use in passenger engines on the broad gauge are seven, eight and nine feet, according to circumstances, the larger diameter being best fitted for the highest speeds. For goods engines on the broad gauge, about six feet is the average. On the narrow gauge, the diameter of the driving wheels now in use for passenger engines is usually six feet, or six feet six inches; and for goods engines, from four feet six inches to five feet.

Four, Six and Eight-wheeled Locomotives.—The engine is supported either on four, six or eight wheels, according to the weight of the machinery and the peculiar views of the engineer as to the comparative merits of the several systems. The advocates of six-wheeled locomotives allege their greater safety in case of fracture of the driving axle—an accident which in a four-wheeled engine is generally, though not always, attended with serious consequences. The four-wheeled engine, from the greater over-hang of the machinery before and behind the extreme points of support, appears to be more liable to pitch or plunge on the rails at high velocities, which is prejudicial, both as regards safety and economy of wear. This tendency may, however, be considerably modified by a skilful distribution of the weight of the engine, with reference to the positions of the wheels, and by the position of the drag-link which connects the train to the engine, &c. The opponents of the six-wheeled locomotive, on the other hand, assert that such a degree of accuracy in the rails and wheels as will permit the weight of the engine, even on level ground, to be supported at six points is practically unattainable; and that, consequently, the parts are exposed to severe strains by being frequently left unsupported. It is well known that six-wheeled engines will plunge violently on the rails if too much of the weight be thrown upon the centre pair of wheels, by screwing them down too tight. To this may be added the increased friction and lateral wear of the rails, caused by six or eight-wheeled locomotives whilst passing curves on the line; but it seems probable that those engines will cause less wear on the *surface* of the rails, as the adhesion necessary for overcoming the resistance of the train is, in their case, spread over a larger surface.

It is found that, under ordinary circumstances, the adhesion of two wheels is sufficient for drawing the train, unless it be a very heavy one; but if the proportion of the weight which is supported by the driving wheels be inconsiderable in comparison with the whole weight of the train, or should the rails be in that slippery state between wet and dry, technically called *greasy* or *dirty*, it will happen that the wheels, in place of progressing along the surface of the rails, will merely revolve in their sockets, and *slip along* the rails without *biting*, or *adhering to* their surface. In such a case, the adhesion of the driving wheels on the rails must be increased, either by throwing upon them a larger portion of the weight of the engine; by throwing sand or ashes upon the surface of the rails, to increase the friction; or by connecting, or *coupling*, together all the four or six wheels of the engine by means of outside connecting rods, the wheels in this case having all an equal diameter.

Coupling the Wheels.—The last is the only method which can be permanently or safely adopted, and is now universally practised for increasing the adhesion of the driving wheels for goods engines in which the wheels are made smaller than for passenger engines. It is apparent that when all the wheels are coupled, the whole weight of the engine is employed in creating the requisite friction, or adhesion, on the rails; but it is found impossible, in practice, to get the weight so equally distributed upon the four or six wheels thus coupled, that all the wheels will wear exactly to the same degree in any given period of time; those which have the most weight to support of course wearing the quickest. The consequence is that after working for some time, the wheels, from the unequal wearing of the tires, lose their uniform diameter; but as they must still all make the same number of revolutions, it follows that the smaller wheels *slip*, to a certain extent, along the rails in order to keep pace with the revolutions of the larger ones, or *vice versâ;* the small wheels either slipping forward, or the larger wheel slipping backwards. Hence a considerable loss of power is the result, should the wheels get out of order.

The rapid reciprocations of the coupling rods are considered objectionable at very high speeds, although coupled engines are sometimes employed at a speed of forty miles an hour. They are, however, inapplicable for those high velocities; since, in the case of express engines, which invariably use large driving wheels to diminish the number of reciprocations of the piston, such a power of traction as would necessitate the coupling of the wheels cannot be obtained.

When the wheels are not coupled, the temptation to throw a very large proportion of the weight of the engine upon the driving wheels is great; but this practice, especially in the case of six-wheeled engines, where the driving wheels are always the central pair, must be avoided as one equally objectionable, both as regards the safety of the train and the security of the permanent way. For although by this means a sufficient adhesion may be obtained, a dangerous degree of motion (both lateral and longitudinal) is generated in the engine, which oscillates and plunges, more or less, upon the driving wheels as a fulcrum. This action is, as may be supposed, highly injurious to the rails, which are bent and displaced by lateral blows from the wheels, and the cast-iron chairs are also sometimes broken by it. When the axles of the wheels, by wearing the brass bushes, acquire lateral play in their bearings, the rocking or oscillating motion is dangerously increased.

Adhering Weight of the Wheels.—There are three considerations which should regulate the amount of adhering weight placed upon the driving wheels of an engine. The first is the power of the cylinders to overcome the adhesion, in proportion to the diameter of the driving wheels; the second is the total weight of the engine; and the third, the construction and strength of the road. The amount of insistent weight to ensure adhesion of the wheels is also dependant upon the weight of the train to be put in motion; the good or bad gradients of the line; and the favourable or unfavourable state of the surface of the rails. Thus it is found in practice that the adhesion varies from one-sixth to one-thirtieth of the insistent weight, according to the state of the rails alone. The same engine, therefore, with an insistent weight of, we will suppose, ten tons on the driving wheels, may at one time exert a tractile force of 3733 lbs. without slipping, while at another time it could not exert more than 746 lbs. The latter is a very extreme case, however, and in practice an adhesion equal to one-sixth of the insistent weight may be generally relied upon.

Resistances to Railway Trains.—This leads us to the consideration of the very important and difficult subject of the various resistances to which railway engines and trains are exposed. The resistance of the blast pipe, as will presently be seen, occasions a serious loss of power in the locomotive, varying from about ten to thirty per cent., according to the speed, and to the construction of the boiler. The other principal sources of resistance are: the friction of the moving parts of the engine, and of the axles of the carriages; the resistance of the air against the front and along the sides of the train; and the effect of gravity on inclines. In a paper upon this subject, read by Mr. Wyndham Harding before the Institution of Civil Engineers, the author states that he had collected from amongst a very large number of the best experiments made by various parties, and by different methods, those experiments only which came under the following conditions, viz.:—That they were conducted at a uniform velocity maintained on a calm day; that the experiments took place on a uniform inclination, and upon a line free from sharp curves; and that the rails and carriages were in good working order. After a careful examination of those experiments, Mr. Harding was led to adopt Mr. John Scott Russel's formula, as applicable to passenger trains under the above conditions, and in this view he is seconded by Mr. R. Stephenson, Mr. Bidder, and other sound practical engineers. In Mr. Russel's formula the whole resistances to the train arising from friction are taken at 6 lbs. per ton; the number of tons weight of the train = T; the velocity of the train in miles per hour = V; the resistance per square foot of frontage, at one mile per hour = ·0025 lbs.; the resistance from concussions, in lbs. per ton of the load, at ten miles per hour and above, = $\frac{V}{3}$; the number of square feet in the frontage of the train = N. The total resistance in lbs. per ton of the weight of the train will then be expressed by the following formula—

$$6 \times \frac{V}{3} \times \frac{(V^2 \times \cdot 0025 \times N)}{T}.$$

The average resistances by actual experiment are stated to be rather less than those found by the formula, at low speeds; but at higher velocities the agreement is said to be very close.

In order to compare with results obtained from the formula just given, we add some interesting experiments on the resistances of engines and trains on the Great Western Railway, undertaken by Messrs. Brunel and Gooch. Their results were obtained from a dynamometer of very large size, with a traction spring six feet long, in connection with arrangements for measuring the speeds, and the resistance and direction of the wind. A mile of perfectly straight and level road was chosen, and a train consisting of nine first and second class carriages, in addition to the dynamometer carriage (taken indiscriminately from the Company's stock), was loaded with iron so as to represent an average passenger train of 100 tons. Indicator diagrams were also taken from the cylinder of the engine at four points during each experiment, as shown in the accompanying Table. It will be borne in mind that the resistance measured by the indicator includes the whole friction of the engine, while the dynamometer experiments show the resistances of the carriages only. Also, that the resistances in the last column are calculated from a formula deduced from experiments made on the *narrow guage*, whilst the second and third columns refer to the *broad guage*.

TABLE OF RESISTANCES WITH A TRAIN WEIGHING ONE HUNDRED TONS.

Speed in miles per hour.	Total resistance of Engine and Train by the Indicator.	Resistance of the Carriages, by Dynamometer.	Calculated resistance to the same Train by the preceding formula.
Miles.	Lbs.	Lbs.	Lbs.
13	9·04	7·67	10·58
20	10·6	9·0	13·13
45	21·9	15·14	24·04
58	25·8	21·5	29·81
61	Not obtained.	22·27	31·57
TABLE OF RESISTANCES WITH A TRAIN WEIGHING FIFTY TONS.			
20	13·0	9·8	13·66
43	19·3	16·16	25·91
51 {Weather wet and stormy.}	27·0	20·53	30·24
62	24·6	22·51	37·46

These experiments indicate a considerably less resistance to the carriages on the broad than on the narrow guage, which is thus explained by Mr. Gooch: "In the first place, larger wheels are used, thereby saving something under the head of friction. Next, the atmospheric resistance, forming a very important element in the whole, is felt not only on the frontage of the train, but much more on the side, acting obliquely at an angle which varies with the direction and force of the wind, and the speed of the train. In several experiments will be found instances in which the angle of atmospheric resistance was as much as twenty to thirty degrees on the side of the train, making its retarding force nearly as much, per square foot of surface, on the side of the train as on the frontage, and considerably more in *total amount*, owing to the extent of surface being thirty or forty times greater, and presenting a very broken surface; and it must not be forgotten that the side pressure produces an injurious effect by forcing the flanges of the wheels against the opposite rails, which the frontage pressure does not."

"The length of the train in proportion to its weight, therefore, becomes a very important consideration, and will account for some of the advantage obtained on the broad guage; for, with a train weighing 100 tons, the side pressure exposed to the atmospheric resistance, with carriages as at present constructed on the narrow guage, is fully fifty per cent. more than it is on the broad gauge. Another element in the resistance to railway trains is the effect of the oscillation and unsteady motion of the train, which has been estimated by Mr. Harding to equal one-third of the velocity in lbs. per ton. This is a very large amount, and certainly much more than is experienced on the broad gauge, where the carriages are larger and on a wider base, and therefore running with greater steadiness."

Mr. Gooch adds, that he has some doubts whether Mr. Russell's formula does not give too high a resistance *even* for the narrow gauge.

We subjoin the following Table on the resistance of trains, under different circumstances, on the Great Western Railway.

TABLE OF RESISTANCES TO RAILWAY TRAINS.

Weight of Train including Engine & Tender.	Length of Line run over.	Speed in Miles per hour.	Total resistance of Engine and Train per ton weight.	Remarks on the Weather.
Tons.	Miles.	Miles.	Lbs.	
184	50	55	28·8	Weather fine.
144	20	54½	24·7	Do.
144	50	56	27·1	Do. rails dirty.
144	20	61	30·5	Do.
94	50	67	33·6	Stormy and wet.
115	50	55	23·1	Fine.
105	50	57	26·1	Fine, with a strong side wind.

Resistance of the Air to Railway Trains.—The resistance of the air is the most serious retarding force to which a train is subjected at high velocities. It is well known that trains left unheeded on a railway have been propelled to considerable distances by the impulse of the wind alone; and that engines, if not sufficiently powerful, may be brought to a standstill by the effect of an adverse or side wind upon the train. The law that the resistance of the air increases as the square of the velocity, within moderate limits, has long been promulgated; but at the higher rates of velocity which are now attained by the locomotive, this law becomes very unsettled, and can by no means be relied upon as affording accurate results. It is also known that a thin plate, passing rapidly through the air, experiences much greater resistance than a prismatic body presenting an equal surface in front; the amount of resistance diminishing according as the prism is elongated, within moderate limits. It has also been demonstrated, by the experiments of M. Thibault, that if two square surfaces be placed so that one shall precisely screen the other, and at a distance apart equal to one of its sides, the resistance against the screened surface will be seven-tenths of that against the surface in front. Hence, in an ordinary train of railway carriages, in which each is separated from the other by an interval of about 2 feet, it has been calculated that the resistance from the air due to each of those intervals is the same as if each separate carriage of the train had exposed a direct frontage surface of one square foot, independently of the additional resistance offered by the rims and spokes of their wheels in rapid revolution. This, for a pair of three feet railway wheels, may be taken at a resisting surface of 4½ square feet, so that each separate carriage may be assumed to expose a direct surface of about 10 square feet to the resistance of the air on a calm day. Should there be wind blowing, this will, of course, affect the resisting surface of the train, by impinging on the sides of the carriages. Experiments have been made to ascertain the retarding effect of the intervals between the carriages, by filling in the open spaces at the sides and top of the train with leather hoods, which are stated to have caused a very considerable reduction in the atmospheric resistance, as measured by the dynamometer.

The actual resistance of the air corresponding to a given velocity of motion has not yet been satisfactorily determined. Smeaton's experiments on this subject were for a long time received as *standard*, but are now believed to be incorrect, showing considerably too high a resistance. The results of Smeaton's experiments are exhibited in the annexed table.

Velocity of the wind in miles per hour.	Perpendicular force on 1 sq. foot in lbs. avoirdupois.	Common appellation of the force of each wind.
1	·005	Hardly perceptible.
4	·079	Gentle pleasant wind.
5	·125	
10	·492	Pleasant brisk gale.
15	1·107	
20	1·968	Very brisk.
25	3·075	
30	4·429	High wind.
35	6·027	
40	7·875	Very high.
50	12·300	A storm.
60	17·715	A great storm.
80	31·490	A hurricane.
100	49·200	Violent hurricane.

The experiments of M. de Pambour, on the other hand, show the atmospheric resistance as follows:—

Velocity of Air in Miles per hour.	Resistance of Air in lbs. per Square Foot of Surface.
5	·07
10	·27
15	·60
20	1·07
25	1·63
30	2·42
35	3·29
40	4·30
45	5·44
50	6·72

It follows as a consequence of the resistance of the air increasing as the square of the velocity, that a limit is thus placed to the speed at which a train can be propelled by a constant force. This is constantly exemplified in practice during the descent of inclined planes, on commencing which the steam is usually shut off from the cylinder (if the incline be a steep one), and the train, committed to its own gravity, runs down with a constantly increasing velocity, until the speed reaches a certain limit, when it then continues uniform to the bottom of the incline. From the same cause, the rate of acceleration during the descent is by no means uniform; following, at the commencement, the law of falling bodies in general, but rapidly diminishing as the speed increases.

Resistance from the Friction of the Machinery.—Having now briefly considered the various resistances to which a railway train as a whole is subjected, it will be necessary to distinguish how much of this total resistance is due to the friction of the machinery. The measure of the friction of any machine, such as an unloaded locomotive engine, is manifestly the smallest force necessary to continue motion throughout the parts of the machine. This may be divided into two portions, viz., that due to the movement of the working parts, and that arising from the weight of the engine considered as a carriage; the friction in the latter case varying in the direct ratio of the weight.

In order to determine the amount of this resistance, experiments have been made upon locomotives in three different ways:—First, by observing the lowest pressure of steam upon the piston at which the motion of the engine, once produced, can be continued; 2ndly, by the direct pull upon a spring balance, or dynamometer; and, 3rdly by the circumstances of their spontaneous descent down an inclined plane, and subsequent stoppage on a level. By the first of those methods, namely, by calculating the effective power in the cylinder, as shown by the indicator, the "Ixion," broad gauge engine, and tender, weighing together about 42 tons, indicated a resistance of 550 lbs., at a speed of 10 miles an hour, this resistance being increased to 789 lbs. at 20 miles an hour, and to 1239 lbs. at 50 miles an hour. The "Ixion" is a six-wheeled engine, with wheels uncoupled, and cylinders $15\frac{3}{4}$ inches diameter and 20 inches stroke. It is found in practice, as might have been anticipated, that the greater the size of the engine, the less is its comparative friction. Thus, in the case of the "Great Britain," broad gauge engine, with cylinders of 18 inches diameter, and supported on eight wheels, the resistance due to the engine and tender, moving at 13 miles an hour, was found to be only 615 lbs.; the weight of the engine and tender being together about 50 tons, and drawing after them a train weighing 100 tons. In this experiment, the dynamometer resistance due to the train was subtracted from the whole power of the engine, as measured in the cylinder; and the engine resistance thus found may be further divided into, say 383 lbs. for the resistance of the engine and tender considered as carriages, and 232 lbs. for the estimated friction of the working parts of the engine. At a speed of 20 miles an hour, under similar circumstances, the sum of the "Great Britain's" resistances amounted to 770 lbs.; viz., 465 lbs. for the friction of the wheels, and 305 lbs. for the moving parts of the engine; at a speed of 45 miles an hour, the sum of the resistances amounted to 1385 lbs., or 726 lbs. for the wheels, and 659 for the moving parts; and at 57 miles an hour, to 1775 lbs., or 1000 lbs. for the wheels, and 775 lbs. for the moving parts of the engine. For the same engine the friction increases nearly as the load; and it will be diminished according as the parts are in good working order, and the engine properly balanced upon its wheels.

Retarding Effect of Gravity.—So long as the train continues running on a level, the only effect of gravity is to cause friction on the axles of the wheels, and at their points of contact with the rails, but when an inclined plane is entered, the case becomes widely different. The gravity of the body, or system of bodies, is then resolved into two forces; the one perpendicular to the surface of the plane; causing pressure; and the other, acting along the surface of the plane, or parallel to it, tending to slide or roll the body to the bottom of the declivity. The portion of the weight which thus tends to drag the body down the plane is easily calculated, when we know the total weight and the slope; the weight being simply multiplied by the fraction which expresses the inclination. Thus, on a plane which rises one foot in a hundred, expressed by the fraction $\frac{1}{100}$. the gravity of one ton, or 2240 lbs., is $\frac{2240}{100} = 22·40$ lbs.; by which the weight of the whole train in tons, including engine and tender, must be multiplied to find the total gravity in pounds.

Tractive Power of Locomotives.—With regard to the tractive power of narrow gauge engines of the present day, we may state that, according to the experiments tried for the Gauge Commissioners, an average velocity of 49 miles an hour was obtained over 8 or 9 miles with a load of 80 tons, exclusive of engine and tender; and over 43 miles, an average velocity of 44·3 miles an hour was obtained. More recent experiments establish the fact that the best passenger engines on the narrow gauge are now capable of propelling a train of 180 tons, including engine and tender, at a rate of 40 miles an hour, on a level; and some engines of still greater power, supported on eight wheels, are now being manufactured. The most powerful engines now in use on the narrow gauge, will propel a train of about 100 tons (their own weight inclusive), at a rate of 60 miles an hour, on a level.

According to the experiments of the Gauge Commissioners, a goods engine on the narrow gauge propelled a train of 400 tons (exclusive of its own weight), at an average velocity of 19 miles an hour, having attained a maximum of 28 miles an hour. The most powerful goods engines now in use on the narrow gauge are calculated to propel a train of 600 tons (including the engine and tender), along a level at a speed of 20 miles an hour; but such unwieldy goods trains are never resorted to, as they are very destructive to waggons and couplings, and most inconvenient at the stations.

On the broad gauge, on the other hand, the most powerful passenger engines are capable of propelling a train of about 230 tons at a speed of 40 miles an hour, on a level; or about 170 tons at 60 miles an hour, the weight of the engine and tender being included in each case. The most powerful goods engines, on the broad guage, will propel a train of 700 to 750 tons, including the weight of the engine and tender, at a speed of 20 miles an hour, when the waggons are in good order.

The following are the average speeds worked on the broad gauge, between London and Exeter, 194 miles; and on the narrow guage between London and Liverpool, 201 miles: On the Great Western Railway, the average speed of the express trains (including a stoppage of ten minutes for refreshments at Swindon), is 43·25 miles per hour, and of the other trains, excepting the 3d class trains, 28·5 miles per hour. On the North Western, the express trains have an average speed of 33·50 miles per hour (no stoppage being included), and the other trains on the same line, excepting third class, 24·17 miles per hour.

The Blast Pipe.—The blast pipe, through which the waste steam escapes into the chimney, acts a most important part in the economy of the locomotive, by supplying an artificial draught to the fires, without which they would burn very languidly, or indeed hardly burn at all. The effect, of course, varies with the greater or less frequency of the respirations or blasts, corresponding to the velocity of the piston, so that the fire is urged more and more according as the speed, and the consequent demand for steam, is increased. During a temporary stoppage, or whilst descending an inclined plane, when the steam power is not so much required, the evaporation from the boiler rapidly sinks to one-sixth or even one-eighth part of what it was previously.

The action of the blast pipe will be readily understood from the following considerations. If we suppose that the waste steam, after having fulfilled its office in urging the piston to the end of the stroke, were to escape into the atmosphere by an orifice of an equal area with the cylinder, then the velocity of the issuing steam would be simply that due to the excess of pressure in the cylinder above the pressure of the atmosphere. But if the velocity of the piston and the vaporization of the boiler remain the same, whilst the aperture through which the waste steam escapes is diminished, it is then evident that the issuing velocity of the steam must be increased in an inverse ratio with the diminution of the orifice, in order to correspond with the same velocity of piston. If, for instance, the area of the blast pipe aperture be reduced to half of what it was previously, the issuing velocity of the steam will then be nearly doubled; and, from the principles of the efflux of fluids, a nearly quadruple pressure will be generated in the blast pipe in order to afford this double velocity. Hence, the pressure in the blast pipe would vary nearly in the inverse ratio of the square of the orifice, provided the flow of waste steam from the cylinders was constant, which it is not: the alternate respirations of the two cylinders producing in the blast pipe a *mean pressure* varying inversely, in a certain proportion, with the area of the orifice.

We have already mentioned that the pressure of steam in the blast pipe is necessary for producing an artificial draught in the chimney to blow the fires. This effect is caused by the rapid issuing of the high pressure steam, which acts both mechanically in giving an upward impulse to the gaseous contents of the chimney, and chemically by condensation; the result being a partial vacuum, into which the external air rushes through the grate bars. The vacuum thus obtained will raise a column of water from two to ten inches, according to the numbers and force of the respirations. The assistance which is thus lent to the boiler by the blast pipe is not obtained, however, without a corresponding and a very serious loss in the power of the engine. This is due to the resistance offered to the piston by the confinement of the steam behind it, owing to the contracted orifice through which it is forced to escape; technically called the *back pressure.* The magnitude of this unavoidable evil may be estimated when we state that at high velocities one-third part of the power of the engine is said to be absorbed by the action of the blast pipe. It thus becomes an important and interesting question to determine how far it is expedient to make use of this expensive auxiliary, and what area of orifice is necessary for the draught, corresponding to a given evaporative power in the boiler, and a given velocity of the engine. Into this question, however, it is unnecessary for practical purposes that we should here enter minutely; since it has now become almost universally the practice to form the blast pipe with a variable orifice, which may be adjusted at will, to the speed of the engine; for it is to be observed, that the pressure against the piston due to the contraction of the blast pipe, varies nearly in the direct ratio of the velocity of the engine; and that by simply increasing the area of the orifice the resistance arising from this cause may be diminished at pleasure. It should also be remembered that the area of the blast pipe must bear a due relation to the other proportions of the boiler, more especially to that which subsists between the sectional area through the tubes, and the grate bar surface. For, if the attempt be made to increase the evaporating power of the boiler by merely elongating the tubes, and thus increasing their heating surface, without providing a corresponding increase in their sectional area for draught, it is apparent that any increase to the heat generated in the furnace (measured by the quantity of oxygen passing through the grate bars), can be obtained only by quickening the draught; or, in other words, by contracting the orifice of the blast pipe. In such a case, the increase to the resistance on the piston, or back pressure, goes far to counterbalance the additional steam power obtained from the boiler.

The following table shows the proportions existing between the fire bar surface and the sectional area through the tubes, with the proportionate diameter of the blast pipe for each, in some engines recently constructed for broad and narrow gauge lines:—

DESCRIPTION OF ENGINES.	Area of Grate.		Area through Tubes.		Proportion of Area of Tubes to Grate.	Proportion of Blast Pipe to Cylinder.	Area of Cylinder.
	Sq. ft.	In.	Sq. ft.	In			Sq. inches.
Great Western Eight Wheeled Engines,......................	21	0	5	3	$\frac{1}{3·9}$	$\frac{1}{13}$	254·4
South Western Engines, ...	10	4	2	7	$\frac{1}{4·0}$	$\frac{1}{14}$	154·9
Grand Junction Do.,............	10	0	2	3	$\frac{1}{4·3}$	$\frac{1}{14·5}$	154·9
Engine 'A,' used at York,.....	9	7	1	7	$\frac{1}{5·7}$	$\frac{1}{17·1}$	176·7

In the case of an engine whose blast pipe had one-fifteenth part of the area of the cylinder, the driving wheel being 7 feet diameter, and the steam cut off at 13⅝ inches of the whole stroke of 18 inches, the loss from the action of the blast pipe at different velocities was as follows:—

At a rate of	7	miles per hour, the driving wheels making	28	revols. per min.	=	2·1	per cent.
—	20	—	—	80	—	=11·6	—
—	40	—	—	160	—	=15·8	—
—	60	—	—	240	—	=21·0	—

We believe the usual proportion for the area of the blast pipe, in the narrow gauge engines, is from one-tenth to one-twelfth of the area of the cylinders.

Vaporization of Locomotive Boilers.—The pressure to which the steam in the boiler is now usually limited by the spring of the safety valve lever varies from 80 to 100 lbs. on the square inch, above the pressure of the atmosphere; this pressure being gradually reduced a little according to the age of the boiler and the supposed wear of the plates and tubes. We have already explained that steam in contact with water always contains the same total amount of heat (latent and sensible) whatever be its tension; hence it follows that the same quantity of heat, or the same weight of fuel, will evaporate an equal quantity of water, independently of the pressure under which vaporization takes place. The vaporization resulting from the combustion of a given weight of fuel is thus theoretically the same at all pressures; but if we consider simply the evaporating power of the boiler, or the quantity of water actually boiled away, without reference to the consumption of fuel, the comparison will be generally in favour of the higher pressure. We have seen that the velocity of the engine, from increasing the respirations of the blast-pipe, influences the vaporization of the boiler in a notable degree. This has been found by experiment to vary nearly in the ratio of the fourth root of the velocity.

The relative value of heating surface in the tubes and the fire-box of a locomotive has long been a disputed point amongst practical engineers. The generally received opinion is, that one square foot of surface in the fire-box will generate as much steam as three feet in the tubes; but M. de Pambour relates an experiment which seems to prove that this relation between the tubes and the fire-box is entirely dependent upon the activity of the combustion and the strength of the draught. The experiment referred to was made in a boiler originally constructed for a locomotive, but used on a stationary engine, in which the two compartments of fire-box and tubes were kept distinct, being separated by an iron partition. This admitted of the evaporation of each compartment being measured separately. "The boiler," he remarks, "was very long, and when the fire was left to itself and the vaporization not abundant, the tubes produced, comparing equal surfaces, an effect considerably less than the fire-box; but by degrees, on the combustion being more excited, and especially when, by means of a blast pipe taken from an adjacent boiler, a more violent jet of steam was applied to the urging of the fire, the effect of the tubes differed less and less from that of the fire box." Our author explains this by supposing that the increased intensity of the draught elongates the flame, and causes it to impinge upon the surface of the tubes throughout their whole length, the tube surface thus becoming more effective in proportion as it absorbs caloric by radiation from the flame, in place of by contact with hot uninflammable gases. We have before observed that we do not subscribe to this doctrine of flame being elongated by intensity of draught.

By taking a mean of the performance of a number of locomotive boilers, we find that each foot of effective heating surface (reckoning fire-box and tube surface together) will evaporate, in round numbers, $\frac{1}{5}$ of a cubic foot of water per hour, at a velocity of 20 miles an hour of the engine; the evaporation corresponding to any other velocity, being deduced from this by multiplying by the fourth root of the ratio between the given velocity and the velocity of 20 miles. When the velocity is *nil*, the effect is reduced to about one-fifth during a temporary stoppage.

We have already remarked that the use of the blast pipe augments the consumption of fuel to five or six times what it was previously, but the corresponding vaporization does not increase in an equal ratio. This is owing to the heating surface of the boiler becoming relatively less in proportion to the heat of the furnace, as the consumption of fuel on the bars is augmented; so that a large portion of heat is wasted by escaping up the chimney. Thus, in an engine at rest, where the blast pipe was not employed, one cubic foot of water was evaporated by 9 lbs. of coke in 4·8 minutes; whereas the use of the blast pipe occasioned the consumption of 11·3 lbs. of coke to evaporate the cubic foot of water, although the time occupied was only ·95 minute. The size of the orifice of the blast pipe does not appear to influence the vaporization so much as might have been expected, at least within certain limits. Hence, if we consider the question of economy of fuel, independently of the time spent in evaporating a given bulk of water from a boiler of given dimensions, the comparison will be doubtless in favour of low speed and slow combustion; but as these are incompatible with the proper working of a railway, we must be content to sacrifice a certain amount of economy to rapidity of combustion and increased evaporative power. Could we conveniently increase the size of the boiler, so as to present an increased effective heating surface in the tubes in proportion to the increased intensity of combustion in the furnace, then a high economy of fuel might still be the result; but as this is necessarily confined within narrow limits, it only remains to effect a fair compromise between these opponent forces, and to proportion the boiler with a due regard to the importance of each. Referring to the experiments of M. de Pambour on this subject, we learn that when the surface of the tubes exceeded that of the fire-box three or four times only, the engine consumed as much as 11·65 lbs. of coke per cubic foot of water vaporized, a large portion of heat escaping up the chimney; but when the surface of the tubes was increased to eight or nine times that of the fire-box, the consumption of fuel per cubic foot of water vaporized was then reduced to 9·18 lbs.; and this saving was effected without the vaporization of the boiler per unit of heating surface being diminished. Finally, by augmenting the heating surface of the tubes beyond ten times that of the fire-box, a further reduction in the expenditure of fuel was indeed obtained, but the vaporization of the engine per unit of total heating surface was found to be reduced at the same time.

The experiments of De Pambour show, therefore, that a ratio of 10 to 1 is the most advantageous that can be established between the total heating surface and that of the fire-box, and that about $10\frac{1}{2}$ lbs. of coke is an average consumption for a good boiler per cubic foot of water vaporized.

In the engines on the Great Western Railway, the superficial surface exposed in the tubes is about ten or eleven times that exposed by the fire-box, such engines evaporating from 8 to $9\frac{1}{2}$ lbs. of water with 1 lb. of coke, according to the rapidity of the evaporation, the slowest evaporation with a boiler of a given size producing the best results. The largest engine now in use on the broad gauge evaporates about 300 cubic feet of water per hour when working with full power at a high velocity, and with a blast pipe one-thirteenth of the area of the cylinder; the area of the fire-box being 153 square feet; thus giving about two cubic feet of water per hour for each square foot of fire-box surface. It is an ascertained fact that a pound of coke will evaporate more water when burnt with a moderate blast in a large boiler than when urged to an extreme heat by a rapid current of air. Thus, in the case of the "Great Britain," (already referred to) with 18 inch cylinders, at an average speed of 28 miles per hour, and with a train weighing on an average 85 tons exclusive of engine and tender, the average consumption of coke per ton per mile is 33 lbs.; whilst the "Ixion," with $15\frac{3}{4}$ inch cylinders, running over the same ground at the same speed and with an average train of 80 tons, burns 35 lbs. of coke per ton per mile. The average consumption of all the engines on the Great Western Railway, running from Paddington station at an average speed of 28 miles, and with an average train of 75 tons, exclusive of engine and tender, is 40 lbs. of coke per ton per mile.

The most powerful engines now in use on the narrow gauge evaporate from 200 to 230 cubic feet of water per hour, and others are now being constructed (with eight wheels) which will increase this amount of evaporation.

The following is a common practical rule for finding approximatively the evaporative power of a locomotive boiler in cubic feet of water evaporated per hour:—"Add the surface of the internal fire-box in square feet (deducting the area of fire bars and the fire door) to one-third of the total heating surface of the tubes in square feet—multiply this sum by 4, and divide by 10." To find the quantity of steam furnished to the cylinders per hour—"Multiply the evaporative power of the boiler (that is, the number of cubic feet evaporated per hour) by the pressure of steam in the boiler per square foot, and again by the volume of steam formed at that pressure, as taken from a table of the volume of steam compared to the volume of the water which produced it[*]—divide the last product by the resistance of the piston per square foot, and the result will be the amount of steam in cubic feet."

Loss of Steam at the Safety Valves, &c.—All engines are more or less liable to a source of waste by blowing off steam at the safety valve, the amount of which less depends on the power of the engine in relation to the load, the gradients of the line, stoppages, &c. It is impossible to calculate the amount of this waste with any certainty, as it is also so entirely dependant upon the experience and care of the engine driver, but it has been valued approximatively at $\frac{1}{20}$ of the whole steam produced during the motion of the engine. A more serious loss of heat results from the water which is carried over with the steam, in a state of mechanical suspension, into the cylinders. We do not allude to *priming*, which is a serious evil in itself, but to the constant abstraction of water, in a fine state of subdivision, accompanying each withdrawal of a cylinder-full of steam from the boiler. The water thus wasted has been calculated at more than $\frac{1}{5}$ of the whole quantity dissipated from the boiler; but the amount of fuel expended in raising this water to the temperature of the steam must not be confounded with what would have been required for actually changing it into steam, no additional heat having yet become latent in it.

Expansion in Locomotives.—Not many years have elapsed since the only degree of expansion considered necessary for the locomotive was obtained by giving a small amount of lap to the slide valves,—and even this concession was made rather with the view of assisting the exhaustion, than of deriving much advantage from the principle of expansion. Recently, however, these engines have been so constructed and arranged as to carry out this important principle to the utmost extent to which it can be advantageously used. This has been effected by enlarging the boilers and cylinders, and by so arranging the valve gear, as to make it capable of nicely balancing the power of the engine, in proportion to the load and the speed, by varying the degree of expansion in the cylinder

In the "Great Britain," broad gauge engine, for instance, the

Diameter of the cylinder is 18 inches. Stroke, 24 inches.
Diam. of driving wheels, 8 feet.
Lap of the slide, outside 1 inch; inside $\frac{1}{16}$ inch.
Area of steam port, 13×2 inches; or 26 square inches.
Lead of the slide, $\frac{1}{4}$ inch full.

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[*] See Table XI., Appendix.

This engine makes use of six different degrees of expansive action, according to the power required, as follows:—

No.	Steam cut off at	Exhaust begins at	Compression begins at
1	18 inches.	22 inches.	21⅜ inches.
2	16⅞ —	21 —	20¾ —
3	14⅜ —	20¼ —	19¾ —
4	2 —	18¾ —	18⅜ —
5	9 —	17¼ —	16⅞ —
6	5⅝ —	15½ —	14⅝ —

In the example we have given, it is the usual practice to make use of the degree of expansion marked No. 1 for starting a heavy train; No. 2 would be employed whilst running at a velocity of forty to fifty miles per hour; but for speeds higher than this, it is remarkable that the best result is obtained by using No. 3. This curious fact is owing to the resistance offered by the blast pipe (or the back pressure) consequent on the escape of the larger quantity of steam, absorbing more power than is gained by the increased pressure on the steam side of the piston; for, as will be observed in the table, the point at which the steam begins to exhaust varies with each degree of expansion, the exhaustion beginning sooner in proportion as the steam is later of being cut off in the cylinder. Hence, supposing the steam to be cut off at 18 inches from the top, the exhaustion would not commence (according to the table) until the piston had arrived at 2 inches from the bottom of its stroke; whereas, by cutting off at 14⅜ inches, the exhaust begins at 3¾ inches from the end of the stroke; thus allowing the smaller volume of steam corresponding to No. 3 nearly double the time to escape that the larger volume corresponding to No. 1 had. The useful effect obtained by the use of No. 3 at a high velocity is therefore greater than that from Nos. 1 or 2, as demonstrated by the indicator diagrams of this engine.

It follows also as a corollary to our last proposition, that the boiler of a locomotive should possess so much effective heating surface as will permit the steam, after being expanded in the cylinder, to exhaust at a moderate pressure through the blast pipe; for should it be necessary, in order to supply a sharp draught in the furnace, that the expanded steam must be forced through a contracted blast pipe, a large portion of the power gained by expansion will be absorbed by back pressure. This consideration imposes a limit to the advantageous use of expansion in each individual case.

The usual practice is to cut off the steam after the piston has passed through from five-eighths to seven-eighths of the stroke; beyond the limit of five-eighths it has not generally been found advantageous to work locomotive engines expansively.

Performance of Passenger Engines.—We subjoin the results of some experiments recently undertaken on the Midland Railway, to determine the power and speed of two new classes of passenger engines, constructed respectively by Messrs. Sharp Brothers & Co. of Manchester, and Messrs. Wilson & Co. of Leeds. Messrs. Sharp's engines have the following dimensions, viz.:—tubes, 161 in number, 2 inches external diameter and 10 feet long. The total heating surface is 847 square feet, of which the fire-box contains 72 square feet. The cylinder is 16 inches diameter and 20 inches stroke, and the driving wheels are 5 ft. 6 in. diameter. Messrs. Wilson's engines have each 124 tubes, 2 inches external diameter, and 11 feet long. Their total heating surface is 800 square feet, of which the fire-box contains 87 square feet. The driving wheels are 6 feet diameter, and the length of stroke is 20 inches.

The weight of Messrs. Sharp's engine in working order was 21 tons 9 cwt., and the tender, with its water and coke, 12 tons 11 cwt., being together 34 tons. The attached train consisted of nine first and second class carriages, loaded with railway chairs, and gave a weight of 64 tons. The gross load to be drawn was therefore 98 tons. The experiments were made from the Derby station to Masborough station, a distance of 40¼ miles. For the first 20 miles the gradients average a rise of 1 in 330, and during the remaining 20 miles they fall in the same proportion. The weather was very favourable, the rails being dry and the air still. As the steam had been up for two hours before starting, the water in the tender was almost at the boiling point when the experiments began. Messrs. Sharp's engine ran the trip at the rate of 49 miles an hour, the first 18 miles (up 1 in 330) being accomplished at the average speed of 43 miles per hour. The total consumption of coke was 16 cwt., or 44·8 lbs. per mile. The total quantity of water evaporated was 10,290 lbs., or 257·2 lbs. per mile; which gives 5·7 lbs. of water per lb. of coke, or about 11 lbs. coke for each cubic foot of water vaporized.

The next trial was made with Messrs. Wilson's engine, the weight of the engine, including water and coke, being 24 tons 1 cwt., and the tender with its water and coke 15 tons 13 cwt., making together 39 tons 14 cwt; or, including the train, 103 tons 14 cwt. This engine performed the trip at the speed of nearly 52 miles per hour, the first 18 miles (up 1 in 330) being passed at the average speed of nearly 47 miles per hour. The total amount of coke consumed was 13 cwt., or 36·4 lbs. per mile. The water evaporated is not stated.

The next trial was made with seventeen loaded carriages, the gross load of the train being 101 tons. Messrs. Sharp's engine consumed during the trip 16 cwt. of coke as before, or 44·8 lbs. per mile; the water evaporated being, this time, 10,840 lbs., or 271 lbs. per mile; showing, therefore, 6 lbs. of water vaporized per lb. of coke, or about 10½ lbs. of coke for each cubic foot of water.

Messrs. Wilson's engine, under the same circumstances, consumed 12 cwt. of coke, or 33·6 lbs. per mile; the total water evaporated being 10·116 lbs., or 252·9 lbs. per mile; equal to 7·5 lbs. of water per lb. of coke, or about 8 lbs. of coke per cubic foot of water.

APPENDIX

CONTAINING

VARIOUS USEFUL TABLES REFERRED TO IN THE PRECEDING TREATISE.

TABLE I.—DIMENSIONS OF MARINE STEAM ENGINES,

AS CONSTRUCTED BY WILLIAM FAIRBAIRN AND Co., LONDON.

DESCRIPTION OF ENGINES.		DIRECT ACTION.			SIDE LEVER.		OSCILLATING.	SIDE LEVER.			OSCILLATING.	
NOMINAL POWER OF SINGLE ENGINE, ...		280 H.P	235 H.P	150 H.P	50 H.P.	40 H.P.	45 H.P.	25 H.P.	20 H.P.	15 H.P.	10 H.P.	5 H. P.
Diameter of cylinder, ...	in inches,	88	80½	65½	40¾	37	39	30	27⅜	23½	15¼	19⅜
Length of stroke, ...	in feet and inches,	5 „ 9	5 „ 9	5 „ 3	3 „ 6	3 „ 0	3 „ 0	2 „ 6	2 „ 3	2 „ 3	1 „ 6	2 „ 0
Piston rod, diameter, ...	in inches,	8¾	8⅜	6¾	4 3/16	3⅝	4⅛	3	2¾	2½	2⅜	2
cone, diam. at top, ...	“	10	10½	7¼	4⅝	4	4¼	3⅛	3¼	2¾	2½	2
“ diam. at bottom, ...	“	10⅜	11¾	8⅝	5¼	4½	5	3⅝	3¾	3¼	3	2¾
“ length, ...	“	12	13	10⅜	7½	7½	7⅜	7	6⅝	6	5	4
“ cutter hole, dimensions, ...	“	Screw	9×2	Screw	2¾×1 3/16	2⅝×⅞	3½×1 3/16	2¾×¾	2½×¾	1¾×½	2¼×½	1¾×½
Cross head to piston rod, necks, diam.	“	8¾	8¼	7⅝	4	3½		2½	2½	2¼		
necks, length of bearing, ...	“	8¾	8¼	7	4½	4		3⅛	3	2¾		
boss, diam. over, ...	“	16	16¼	11⅜	7¾	7		5⅞	5⅝	5¼		
“ hole for head of rod, diam.	“	8¾ to 8¼	8⅛	6½	4	3½		2¾	3⅛	2¼		
“ depth, total, ...	“	23¼	20½	17⅛	10⅜	9⅛		7½	7	6⅜		
“ cutter hole, dimensions, ...	“	3¾×1⅞	Brass nuts on head.		3×¾	3×¾		2¼×⅝	2¾×⅝	2¼×7/16		
blade, depth at centre boss, ...	“				9⅝	8½		6⅝	6½	5½		
“ depth at ends, ...	“				4½	4⅛		3¾	3	2¾		
“ thickness at boss, ...	“				2¾	2¾		2	1⅞	1⅝		
“ thickness at ends, ...	“				2⅝	2½		2	1⅞	1⅝		
Connecting rod, diam. in middle, ...	“	10¾	10½	8¼	5⅞	5		3¾	3½	3⅝		
diam. at ends, ...	“	9¾	9½	7⅜	4¾	4⅛		3	3	2¾		
length from centre of crank pin to centre of cross tail,	in ft. and in.	8 „ 10⅛	8 „ 7½	7 „ 5	7 „ 0	6 „ 9¼		5 „ 7½	4 „ 1¾	4 „ 10¾		
depth at head, within strap, ...	in inches,	14¼	13	10⅝	6⅜	5¾		4¾	4½	3⅝		
breadth of strap, ...	“	9¾	9⅜	8⅞	5	4		3¼	3¼	3⅜		
strap, thickness at crown, ...	“	5	3⅞	3¾	2	1⅞		1⅝	1½	1⅛		
“ thickness at thinnest part, ...	“	3¾	2⅛	2⅞	1 11/16	1⅜		1¼	1⅛	⅞		
“ cutter hole, dimensions, ...	“	13×2½	11×2⅝	9×2¼	5×1⅛	4½×1		3¼×⅞	3¾×¾	3×⅞		
“ metal below cutter hole, ...	“		7¼	5⅝	3½	3		2½	2⅜	2¼		
bearing on crank pin, diam.	“	12	11	9	5½	5	5	4	3⅝	3	3	2½
“ length, ...	“	12½	12	11	7	5⅛	6½	4½	4⅝	4⅝	3½	3
“ thickness of brass at crown, ...	“	1½ to 1⅛	1⅜	1¼	¾	¾	¾	¾	¾	1 1/16	⅝	⅝
Cross tail on connecting rod, necks diam.	“				4 3/16	3⅝		3	3	2¼		
necks, length, ...	“				4½	4½		3½	3⅜	3		
boss, diam over, ...	“				8¼	7½		6½	5¾	5½		
“ hole for connecting rod, ...	“				4½	3¾		3¼	3⅛	2¾		
“ depth, total, ...	“				10½	9¾		7¾	7	6¼		
“ cutter hole, dimensions, ...	“				3¼×⅛	3×¾		2½×⅝	2⅝×⅝	2¼×½		
blade, depth at centre boss, ...	“				10	8¾		6¾	6½	5½		
“ depth at ends, ...	“				4⅞	4⅝		4	3¼	2⅞		
“ thickness, ...	“				2⅝	2¾ to 2⅛		2⅛	2	1⅝		
Cross tail link, diam. over boss, ...	“				6¼	5½		4½	4¼	3⅝		
bearing on side lever, diam.	“				3½	3		2½	2⅜	2¼		
“ length, ...	“				4¾	4½		3⅛	3	3		
strap, breadth, ...	“				3½	3⅜		2⅜	2½	2¼		
“ cutter hole, dimensions, ...	“				3¼×¾	3×¾		2½×⅝	2½×⅝	2⅜×⅝		
Cranks on paddle shaft, large boss, hole for shaft, diam. ...	“	17	16	14	8½	7⅝	7¾	5½	5⅝	4½	4¼	3¼
boss diam. over, ...	“	31	29¾	25½	14¼	13½	13½	10	10¼	9¼	7½	6
“ depth, total, ...	“	13½	14	13	7⅝	6¾	8⅜	5¾	5¾	5	4½	3 3/16
eye for pin, diam. over, ...	“	24	21½	18	9¾	9½	9	7	7¼	6⅜	5½	4½
“ depth, ...	“	11⅝	12	10⅛	6	5½	6¾	4¾	4⅝	3¼	3½	2⅜
web breadth at large boss, ...	“	20½	18½	16¾	9¾	8⅜	8¾	7⅜	6¾	5½	5	4
“ breadth at eye, ...	“	17½	16	13	7⅛	6¼	6½	5⅝	5¼	4¾	3⅜	3½
“ thick at large boss, ...	“	8½	8¾	7½	4¼	4¼	4⅛	2⅝	3	3	2⅝	2¼
“ thick at eye, ...	“	8½	8¾	7⅜	4	4	4⅛	2⅝	2⅝	2½	2½	2¼
Crank pin, diam.	“	12	11	9	5½	5	5	4	3⅝	3	2⅞	2⅜
length of bearing, ...	“	12½	12	11	7	5⅛	6½	4½	4⅝	4⅝	3¼	2¾
cone, diam. at top, ...	“	12 1/16	11¼	9 5/16	5½	5	5¼	4¼	3 5/16	3¼	3	2½
“ diam. at bottom, ...	“	12⅜	11⅝	9⅞	5¾	5¼	5⅝	4½	4⅛	3½	3 3/16	2 11/16
Paddle shaft, necks within board, diam.	“	17	15¾	13½	8½	7½	7½	5½	5½	4½	4¼	3¼
necks, length, ...	“	18	18	18	10⅝	10¾	10	8⅝	8	6¾	5⅞	4
outside bearing, diam.	“	13½	12¼	11	6	5⅝	10	4½	4½	3¾	3½	3¼
“ “ length, ...	“	16¾	14½	13	12	10½	17	8½	7¾	8	3½	3⅛
Intermediate shaft, diam. in middle (necks are same as paddle shaft),	“	18½	17	14½	9⅜	8¼	8¾	5½	6	5¼	4¾	4¼

TABLE I.—*Continued.*

DESCRIPTION OF ENGINES.		DIRECT ACTION.			SIDE LEVER.		OSCILLATING.	SIDE LEVER.			OSCILLATING.	
NOMINAL POWER OF SINGLE ENGINE, ...		280 H.P.	235 H.P.	150 H.P.	50 H.P.	40 H.P.	45 H.P.	25 H.P.	20 H.P.	15 H.P.	10 H.P.	5 H. P.
Side rods, diam. in middle, ...	in inches,				4 1/8	3 7/8		2 7/16	2 3/4	2 5/8		
diam. at ends, ...	"				3 3/8	2 7/8		2 3/16	2 1/4	2		
depth at lower end within strap, ...	"				4 1/4	3 1/2		3 1/8	3	2 3/4		
bearing on side lever, diam.	"				3 1/2	3		2 1/2	2 3/8	2 1/4		
" " length, ...	"				4 5/8	4 3/8		3 1/8	3 1/8	3		
strap, breadth, ...	"				3 1/2	3 1/4		2 1/2	2 7/16	2 1/4		
" thickness at crown, ...	"				1 3/4	1 1/4		1 1/8	1	7/8		
" " at thinnest part, ...	"				1 5/16	15/16		7/8	7/8	5/8		
" cutter hole, dimensions, ...	"				3 3/8 × 3/4	3 × 3/4		2 1/4 × 5/8	2 3/8 × 5/8	2 1/4 × 1/2		
" metal below cutter hole, ...	"				2	2 1/4		1 3/4	1 1/2	1 3/8		
eye on cross head, diam. across, ...	"				8 1/2	7		6	5 5/8	4 3/4		
" thickness at crown, ...	"				2 1/4	1 7/8		1 1/2	1 1/4	1 1/8		
bearing on cross head, diam.	"				4	3 1/2		2 1/2	2 1/2	2 1/4		
" length, ...	"				4 1/2	4		3 1/8	3	2 3/4		
eye for stud to radius rod, ...	"				1 3/8	1 1/4		1	1 1/8	1		
" diam. over, ...	"				4 7/8	4		3	3 1/4	3		
Air pump rod, diam., ...	"	5 3/8	5 13/16	4 7/8	2 3/4	2 5/16	2 3/4	2 1/8	2	1 1/2	2	1 1/4
brass lining, thick, ...	"	1/2	1/2	1/2	3/8	1/4	none.	3/8	3/8	3/16	1/4	none.
cone, diam. at top, ...	"	5 1/2	6 1/8	5	2 7/8	2 1/2	3	2 1/4	2 1/8	1 5/8	2 1/8	1 3/8
" " at bottom, ...	"	6 3/4	6 7/8	5 1/2	3 1/4	3	3 3/8	2 5/8	2 1/2	2 1/4	2 1/2	1 11/16
" length, ...	"	8 1/2	8 5/8	7 1/4	4 1/2	5	5 1/2	4	3 1/2	4 1/2	4	3
" cutter hole, dimensions, ...	"		3 1/4 × 1 1/4	2 3/4 × 1	2 1/2 × 5/8	2 × 9/16	2 3/8 × 5/8	2 × 1/2	1 3/4 × 9/16	1 3/8 × 3/8	1 3/8 × 7/16	1 1/4 × 3/8
Air pump cross head, necks, diam.	"	5 7/16	5	4	2 1/4	2		1 3/4	1 1/2	1 1/2		
necks, length of bearing, ...	"	5 7/16	5	4 1/8	2 3/4	2 1/2		2 1/4	1 3/4	1 3/4		
boss, diam. over, ...	"	11 1/8	10	8 5/8	5 3/4	5		3 3/4	4	3 1/2		
hole for head of rod, at widest part, ...	"	5 7/16	5 1/2	4 1/8	2 1/8	2		2 1/8	1 15/16	1 1/8		
boss, total depth, ...	"	12	11	10	7	6 1/4		5	4 5/8	4 3/8		
" cutter hole, dimensions, ...	"	5 × 1	2 1/2 × 1	3 1/2 × 1	2 1/4 × 5/8	2 1/8 × 1/2		2 1/8 × 3/8	2 × 3/8	1 1/2 × 3/8		
blade, depth at centre boss, ...	"	11 1/2	10 3/8	9 1/2	6 1/4	5 7/8		5	4 3/8	4 3/8		
" depth at ends, ...	"	6 5/8	6	5	3	2 1/2		3	2 1/4	2 1/8		
" thickness, ...	"	3	3 3/16	2 11/16	1 7/8	1 5/8		1 5/8	1 9/16	1 3/8		
Air pump side rods, diam. in middle, ...	"			3 7/8	2 5/16	2		1 5/8	1 5/8	1 5/8		
" " diam. at ends, ...	"			3 1/8	2	1 5/8		1 3/8	1 1/2	1 1/2		
bearings, diam.	"	5 7/16	5	4	2 1/4	2		1 3/4	1 1/2	1 1/2		
" length at lower end, ...	"			4 1/8	3	2 7/8		2 1/8	2 3/8	2		
Main centre, necks, diam.	"				6	5 1/2		4 1/2	4	3 3/4		
necks, length, ...	"				9 1/4	8 1/8		6 3/4	6	6 1/2		
bosses, diam.	"				6 1/2	6		5 1/2	5	4 3/4		
Motion side rods, diam. in middle, ...	"					2 1/8		1 3/4	1 3/4	1 5/8		
diam. at ends, ...	"					1 1/2		1 1/4	1 3/8	1 3/8		
depth at top, within strap, ...	"					2		2	2	2		
strap, breadth, ...	"					1 3/4		1 3/4	1 3/4	1 3/4		
" bearing, diam.	"					1 1/2		1 5/8	1 1/2	1 1/2		
" " length, ...	"					2 1/4		2 1/4	2 3/8	2		
eye on motion shaft, diam.	"					1 1/2		1 1/4	1 3/8	1 1/2		
" bearing, length, ...	"					2 1/4		2	1 1/2	2		
" diam. across, ...	"					3 3/8		3 1/4	3 1/8	3 1/8		
Motion shaft, diam. of necks, ...	"				2 3/4	2 1/4		1 3/4	1 3/4	1 1/4		
length of necks, ...	"				5 1/2	3 1/8		2 7/8	3	3		
cranks to radius rods, centre to centre, ...	"				7 9/16	8 1/2		6 3/16	5 1/8	4 7/8		
studs to do. do. diam. and length, ...	"				1 1/2 × 1 5/8	1 1/4 × 1 1/2		1 × 1 3/16	1 1/8 × 1 1/4	1 1/8 × 1 1/2		
Radius rods, diam. in middle, ...	"	4 3/4	4 3/4	3 3/4	2 1/8	1 5/8		1 1/4	1 3/8	1 3/8		
" diam. at ends, ...	"	4	4	3 1/8	1 1/2	1 1/4		1	1	1 1/8		
" bearings, diam.	"	5	5	4	1 1/2	1 1/4		1 1/4	1 1/4	1 1/8		
Weigh shaft, necks, diam.	"	7		5 1/8	3 1/8	2 3/4	2 3/4	2 1/4	2 3/8	2 1/4	2	1 15/16
" necks, length of bearing, ...	"	9		7	4 1/2	3 1/2	4	3 1/2	4 1/4	3 1/2	2 1/2	2
studs for valve links, diam.	"	3 1/2		3 1/2	1 1/2	1 3/8	1 1/2	1 1/4	1 1/4	1 1/8	1 1/8	1
Starting lever, deep in middle, ...	"	wheel.	wheel.		2 7/8	2 7/8	2 5/8	2 1/2	2 3/8	2 1/8	1 3/4	1 3/4
" thick, ...	"				1 1/4	1 1/4	1 1/4	1 1/8	1	1		1/2
" leverage, ...	"	12 : 1	15 : 1	13 : 1	4 : 1	4 : 1	6 : 1	4 : 1	3 3/4 : 1	3 : 1	4 : 1	6 : 1
Eccentric, diam. to inside of brass ring, ...	"	41 3/8	34 3/8	32	22	19	18 3/4	14 3/8	14 1/2	13	11 1/4	9 1/2
throw, ...	"	20	14	14	9	7 1/2	7 1/2	6 1/4	6 1/4	4	3 1/4	3 5/16
breadth of brass ring, ...	"	4	3 1/2	3 1/2	2	1 3/4	1 11/16	1 3/8	1 3/4	1 3/8	1 3/8	1 1/8
rod, diam. at top, ...	"	6 3/4	3 of 2 1/8	3 of 2	2 1/4	2	2	1 3/8	1 3/4	1 3/4	1 3/8	1 1/8
" " at middle, ...	"		4	3 3/4	2 7/8	2 1/2	2 3/8	1 7/8	2	2 1/8	1 3/8	1 1/8
" " at end next bow, ...	"	5 3/4	3 1/8	3	2 1/8	1 3/4	1 3/4	1 1/4	1 1/2	1 1/2	1 3/8	7/8
" breadth of sliding face beyond the stud, ...	"	3 3/4	3 3/4	3 5/8	2 1/4	2	2	1 1/4	1 1/2	1 5/16	1 1/4	1 1/8
eccentric stud, diam.	"	4	3 3/4	3 1/2	1 3/4	1 5/8	1 5/8	1 1/4	1 5/8	1 1/2	1 1/4	1
Valve cross head, necks, diam.	"	3	2 3/4	2 1/2	1 9/16	1 3/8		1 1/4	1 1/4	1 1/4		
" boss, depth, ...	"	6 1/2		4 7/8	3 1/2	3		2 1/4	2 3/8	2 3/8		
" " diam. over, ...	"	6		4 3/4	3	2 1/2		2	2 1/4	2		
Valve links, breadth, ...	"	3 1/2	2 1/8	2	1 5/16	1 1/2		1 1/16	1 1/4	1 1/8		
Valve rod, diam. at thinnest part, ...	"	3	3	2 3/4	1 3/8	1 1/4	1 1/2	1	1 1/8	1 1/8	1	3/4
Piston, depth in centre, ...	"	12	13	10	7	6 3/4	7	6 1/2	6 1/2	5	4 1/2	4
" at circumference, ...	"	8 1/2	9 3/4	10	7	6 3/4	7	6 1/2	6 1/2	5	4 1/2	4
" metallic packing, ...	"	6	7	7 1/4	5	5	4 3/4	4 1/2	4 5/8	3 1/4	2 3/4	2 1/4
thickness of do.	"	2 1/2	4 1/2	4	2 1/4	2 1/4	2 3/8	1 3/4	1 1/4	1 1/2	1 1/2	1 1/2
Side lever, depth in centre, ...	"				22 1/2	20			16	14 1/2		
" " at ends, ...	"				9 1/2	8 1/2			7 3/8	6 1/2		
breadth across centre boss, ...	"				9 1/4	8 1/8			6 3/4	6 1/2		
" flange at middle, ...	"				5 3/4	5 1/2			4 3/4	4 1/2		
" " at ends, ...	"				4	3 3/4			3 1/2	3 1/8		
thickness of metal, ...	"				1 1/4	1			1	3/4		
Bilge and feed pumps, diam. of plunger, ...	"	9 1/2	9 7/8		3 3/4	3 1/2	3 1/2	3 1/4	2 7/8	2 1/4	1 3/4	1 5/8
length of stroke, ...	"	34 1/2	34 1/2	31 1/2	22 1/2	18	18	15	14	13	11 1/2	10

TABLE II.—DIMENSIONS OF PARTS PECULIAR TO MR. FAIRBAIRN'S DIRECT ACTION ENGINES.

NOMINAL POWER OF SINGLE ENGINE,		280 H.P.	235 H.P.	150 H.P.
Piston rod, diam. of head in crosshead,	in inches,	8⅝ to 8¼	8⅛	6⅞
length of head in boss,	"	20½	20½	17⅛
screw at top end, for brass nut, diam.,	"	none.	5⅝	4¼
" " " length,	"		7½	6
brass nut on head of piston rod, external diam.,	"		9½	6⅞
Crosshead to piston rod, diam. over centre boss,	"	16	16¼	11⅜
depth of boss, total,	"	23¼	20½	17⅛
necks for connecting rod, diam.,	"	8¾	8¼	7⅝
" " length,	"	8¾	8¼	7
necks for hanging motion beams, diam.,	"	6½	6	4¾
" " " length,	"	6½	6½	5½
Connecting rod, bearings on crosshead, diam. each,	"	8¾	8¼	7⅝
fork end on crosshead, depth within straps, each,	"	10¾	9¾	7¾
" straps, breadth, each,	"	6⅝	6¼	5½
" " thickness at crown,	"	4	3¼	3¼
" " " at thinnest part,	"	3	2 9/16	2⅜
" " metal below cutter hole,	"	5¼	4⅞	4½
" brass, thickness at crown,	"	1¼	1⅛	1
" cutter hole, dimensions,	"		8¼×1⅝	7×1⅜
" cutter within strap, breadth,	"		3⅛ to 2⅛	3¼ to 2¼
Air pump crosshead guides, internal diam.,	"	3	2¾	2½
" " length,	"	6⅝	6¼	5
" " diam. over boss,	"	6	5½	4⅜
Air pump side rods or links, at crosshead end, diam. of bearing,		5 7/16	5	4
length of bearing at this end,	"		5	4⅛
depth within strap,	"	Link.	Link.	5
breadth of strap,	"	4	3¾	3¼
cutter hole in do., dimensions,	"	5⅛×1	5×⅞	4×13/16
eye on rocking beam, diam. of bearing,	"	5⅝	5	4
" " " length of do.,	"	5 7/16	5	4⅛
" " " breadth across,	"			8⅛
key for brass, breadth within the eye,	"			1⅝ to 1¼
" " thickness,	"			13/16
Hanging beams for motion, diam. across centre boss,	"	15¾	19	13¾
diam. of eye of boss,	"	7	7¾	6¼
length of bearing of eye,	"	6½	5⅜	6
bearing on piston crosshead, diam.,	"	6½	6	4¾
" " " length,	"	6½	6½	5½
depth of beam at this end within strap,	"	8	7 11/16	6
strap, breadth,	"	5	5	4

NOMINAL POWER OF SINGLE ENGINE,		280 H.P.	235 H.P.	150 H.P.
Hanging beams for motion strap, thickness at crown,	in inches,	3½	3⅝	2⅛
" cutter hole, dimensions,	"	6×1¼	6×1¼	4⅞×1
boss on end of beam at vibrating shaft, diam. over,	"	12	13¾	10
eye in do., diam.,	"	5½	6¼	5½
" breadth,	"	5	5	5
thickness of blade of this beam,	"	2	2¼	1⅞
Vibrating shaft for motion, diam. at middle,	"	6⅛	6⅜	5
necks for radius rods, diam.,	"	5	5	4
bosses for ends of hanging beams, diam.,	"	6	6¼	5½
Rocking beams for working air pump, diam. over centre boss,		18½	22¾	17½
eye in centre boss, diam.,	"	8	7½	7½
" " depth,	"	11	11	6½
breadth of beam on either side of centre boss,	"	18	17½	13
thickness, do.,	"	2⅜	2⅜	2
bearing on hanging motion beams, diam.,	"	6½	6	4⅞
" " " " length,	"	6½	6½	5½
depth of beam at this end within strap,	"	7⅞	7¾	6
strap, breadth,	"	5	5	4
" thickness at crown,	"	3⅛	3⅝	2⅛
" cutter hole, dimensions,	"	6½×1¼	6×1¼	5×1
breadth of beam at this end,	"	7⅞	7¾	6
thickness, do.,	"	2⅛	2⅜	1¾
boss at air pump end, diam. over,	"	13⅛	13¼	10½
" eye, diam.,	"	6⅝	6¼	5½
" " depth,	"	5	5	4½
breadth of beam at this end,	"	8½	9⅜	8
thickness, do.,	"	2⅛	2⅜	1⅝
Main centre for rocking beams to air pump, diam. in middle,		9½	9⅛	7
necks in pedestals, diam.,	"	7½	7½	6
" " length,	"	11	11	7½
Main centre for hanging motion beams, diam. in middle,	"		7⅞	6
diam. at ends next necks,	"		6½	4⅞
Upper shaft for radius rods, diam. in middle,	"	6½	6	5⅜
diam. at ends next bosses,	"	5	5	4
diam. of bosses in entablature,	"	6½	6	5½
Vibrating shaft for radius rods, diam. in middle,	"		6⅜	5
diam. at ends next bosses,	"		5	4⅛
Shaft on rocking beams for air pump side rods, diam. in middle,			6¼	5⅜
diam. at ends next necks,	"		5	4¼

TABLE III.—DIMENSIONS OF PARTS PECULIAR TO MR. FAIRBAIRN'S OSCILLATING ENGINES.

NOMINAL POWER OF SINGLE ENGINE,		45 H.P.	10 H.P.	5 H.P.
Piston rod, diam. of head in boss,	in inches,	4	2¼	1 15/16
length, " "	"	11	5	4⅜
diam. over boss,	"	6¾	3⅝	3
cutter hole, dimensions,	"	4×⅞	2¼×½	
diam. of brass bearing on top,	"	5	2⅞	2¼
length " "	"	6½	4¼	3¼
Intermediate shaft crank, diam. of bearing,	"	9	5⅛	4
" " length "	"	6	3¼	2
Air pump connecting rod, length centre to centre,	"	36	18	15
diam. at middle,	"	2 of 1⅞	2¼	1⅝
bearing on crank, diam.,	"	9	5⅛	4
" " length,	"	6	3¼	2

NOMINAL POWER OF SINGLE ENGINE,		45 H.P.	10 H.P.	5 H.P.
Air pump connecting rod,				
bearing on air pump crosshead, diam.,	in inches,	4½	2⅞	1½
" " " length,	"	4	3	1½
Air pump rod, head in boss, diam.,	"	2⅝	1¾	1¼
" " " length,	"	7	3	2¼
diam. over boss,	"	4½	2¾	2⅛
bearing on guide crosshead, diam.,	"	4½	2⅞	1⅝
" " " length,	"	4	2⅞	2⅞
Air pump guide crosshead next boss (round), diam.,	"	3⅞	2¼	1⅝
" " next guides, "	"	2⅜	1⅜	1⅛
Cylinder, rocking neck, external diam.,	"	14½	8⅝	6
" " length of bearing,	"	6 3/16	3½	2 13/16

TABLE IV.—DIMENSIONS OF THE PRINCIPAL PARTS OF MESSRS. MAUDSLAY AND FIELD'S MARINE ENGINES.

NOMINAL POWER OF SINGLE ENGINE, ...		120 H.P.	110 H.P.	100 H.P.	90 H.P.	80 H.P.	70 H.P.	60 H.P.	50 H.P.	40 H.P.	30 H.P.	25 H.P.	20 H.P.	15 H.P.	10 H.P.
Diameter of cylinder, ...	in inches,	57	55½	52½	50	48	46	43	40	36½	32	29½	27	24	20
— piston-rod, ...	"	5½	5¼	5	4⅞	4¾	4½	4¼	4	3½	3¼	3	2¾	2⅜	2
— air pump, ...	"	34	31½	30	28	27½	26	24	23	21	18½	17½	17	15	12
— air pump rod, ...	"	4¼	4	3¾	3½	3¼	3	2⅛	2¾	2½	2¼	2⅛	2	1¾	1¼
— injection cock, ...	"	3½	3⅜	3¼	3¼	3⅛	3	2¾	2½	2¼	2	1¾	1⅝	1½	1¼
— hot water pump, ...	"	7½	7	6½	6	5½	5	4½	4¼	4	3½	3¼	3	2½	2¼
— feed pipe, ...	"	4	3½	3½	3¼	3¼	3	2¾	2½	2½	2¼	2⅛	2	1¾	1½
— steam pipe, ...	"	12	11½	11	10½	10	9¼	8½	7¾	7	6½	6	5¾	5	4
— waste water pipe, ...	"	14	13½	13	12¼	11½	10½	10	9½	9	8	7½	7	6	5
— beam gudgeon, ...	"	9¾	9½	9	8½	8	7½	7	6½	6	5½	5¼	5	4¼	3½
— pins in beam ends, ...	"	5½	5¼	5¼	5	4½	4½	4¼	4	3½	3¼	3	2¾	2⅜	2
— air pump pins in beam,	"	3¼	3⅛	3⅛	3	2⅞	2⅞	2½	2½	2¼	2	1⅞	1¾	1⅝	1¼
— crank pin, ...	"	8	7¾	7⅜	7	6½	6	5½	5	4½	4	3¾	3½	3	2½
— main shaft, ...	"	12½	12	11½	10½	10¼	10	9¼	8½	7½	7	6¾	6¼	5½	4¼
— paddle wheels, in feet, ...	"	23	21	21	19	19	17	17	15	13	13	12	11	11	9
— weigh shaft bearings,	"	3¾	3¾	3½	3¼	3¼	3	2¾	2¾	2¾	2⅝	2½	2½	2¼	2
Stroke of piston, ...	"	72	66	63	60	56	52	48	42	36	36	33	30	30	24
— air pump bucket, ...	"	36	33	31½	30	28	26	24	21	18	18	16½	15	15	12
— feed pump plunger, ...	"	18	16½	16	15	14	13	12	10½	9	9	8	7½	7½	6
Cylinder cross head, depth of boss,	"	17½	17	16	15	14½	14	13	12	10½	9½	9	8	7½	6
— — diameter of boss,	"	12	11	10	9½	9	8¾	8	7½	6¾	6	5½	5	4½	4
— — depth of middle,	"	14	13½	13	12½	11¾	11½	10¼	9½	8½	7½	7	6¼	5½	5
— — thickness, ...	"	4⅛	4	3¾	3½	3½	3¼	3	2¾	2½	2¼	2	1¾	1⅝	1½
Air pump cross head, depth of boss,	"	12½	12	11½	11	10¾	10½	10	9	8	6¾	6½	5½	5	4½
— — diameter of boss,...	"	7½	7¼	6¾	6½	6	5¾	5¼	5⅛	4⅝	4⅛	4	3½	3⅜	2¾
— — depth of middle,	"	9½	9¼	9	8½	8¼	8	7⅝	7	6¼	5¼	5	4½	4	3½
— — thickness, ...	"	2¾	2¾	2½	2⅜	2¼	2¼	2⅛	2	1¾	1½	1⅜	1¼	1⅛	1
Centre to centre of air pump side rods transversely		72	70	68½	67	63	60¾	55¾	53	47½	42¼	39½	37¼	34¼	29½
— — beams, do.	"	83	80	78	72	69	69	63	60	54	48	45	42½	39	33
— — frames, do.	"	46	45	44	42	40	40	34	34	30	27	26	25½	23	21
— — engines, do.	"	130	128	126	112	108	108	100	96	88	84	80	76	72	66
Steam port, length, ...	"	21	20	20	19	19	18½	18½	15	13	11½	11	10	8¾	7½
— — breadth, ...	"	4¾	4¾	4½	4¼	4	4	3	3	2¾	2½	2¼	2	1¾	1½
Foot valve passage, area, ...	"	224	201½	186	159½	147	140	117	96	75	58½	42½	38¾	28	26
Beam, breadth at middle, ...	"	39	38	36	35	34	33	29	28	25	23	21	19	18	14
— breadth at ends, ...	"	15½	15	14	12¾	12¼	12	10½	10	8¾	8	7½	6¾	6	5
— thickness, ...	"	2⅝	2½	2½	2½	2⅜	2¼	2	1⅞	1¾	1½	1⅜	1⅛	1⅛	1

TABLE V.—PROPORTIONS OF MARINE ENGINES BY DIFFERENT MAKERS, CORRESPONDING TO A GIVEN WEIGHT OF 300 TONS, WITH NOT LESS THAN 300 TONS OF COALS IN BOXES.

NAMES OF THE ENGINEERS.	DESCRIPTION OF ENGINES.	Total Horse Power, Admiralty Estimate.	Total Horse Power, Contractor's Estimate.	Estimated weight of Engines.	Weight of Paddle Wheels.	Weight of Duplicates, &c.	Total weight of Machinery and Water.	Total weight with Water, per Admiralty H. P.	Diameter of Cylinder.	Length of Stroke.		Strokes per Minute.	Diameter Necks of Paddle Shaft.	Time for making Machinery.	Time for fixing Machinery.	Length of the Engine Room.		Nett length of the Engine.		Length of the Connecting Rod.		Ratio of the Connecting Rod to Crank.	Height of centre of Wheel above Water Line.	
		H. P.	H. P.	Tons.	Tons.	Tons.	Tons.	Cwt.	Inches.	ft.	in.	No.	Inches.	Mos.	Mos.	ft.	in.	ft.	in.	ft.	in.	Ratio to 1.	ft.	in
Boulton & Watt,	Direct.	484	480	140	30	17	300	12·39	82	5	6	20	14	8	2½	56	3	13	3½	8	3	3·0	8	10½
Bury, Curtis & Co.,	...	427	426	140	24	20	298	13·96	77	5	6	20	14½	10	2½	47	8	16	7	14	0	5·09	8	9
Butterley Co.,	...	423	420	135	23	20	297	14·04	77	5	4	20	14	9	2	48	1	12	3	7	6½	2·83	8	9
Caird & Co.,	...	496	500	150	20	10	300	12·09	83	5	6		17	10	2½	56	0	11	11	8	9	3·28	8	9
Fairbairn & Co.,	...	463	460	148	29	20	300	12·96	80¾	5	3	20	14½	7	2	45	10	15	6	6	0½	2·30	8	10
Forrester & Co.,	Oscillating.	460	460	110	26	26	300	13·04	76½	7	6	16	17¼	6	2½	56	3	18	3	13	9	3·66	8	9
Fawcett & Preston,	...	500	500	99	40	12	300	12·0	81	6	9	17	16	6	2	52	3	17	1	12	7	3·77	8	9
Do. Do.	Direct.	444	444	105	37	19	300	13·51	78	5	9	19	15¼	6	2	56	2	14	1½	13	10	4·88	8	10
Maudslay & Field,	Double Cyl. Direct.	456	450	148	26	5	300	13·16	4 of 55½ 2 of 78½	6	0	18½	14¾	5	2	56	2	18	1	12	5½	5·07	8	10
Miller & Ravenhill,	Oscillating.	544	544	105	26	27	300	11·03	85	6	6	17½	16½	9	3	53	0	17	6	13	8	4·20	8	9½
Robert Napier,	Direct.	508	500	125	30	8	300	11·81	84	5	6	20	15	6	1½	56	2	17	7	7	10	2·938		
Penn & Son,	Oscillating.	501	500	117	38	23	292	11·65	82¼	6	0	18½	16	8	3	56	1	13	5	13	2	4·39		
Rennies,	Direct.	501	500	123	35	13	300	11·97	83	5	8	19¼	16¼	8	2	46	9	11	10				8	11¼
John Rigby,	...	515	510	127	33	15	300	11·65	84	5	9	19	18	6	3	48	3	17	1	7	7	2·76	8	9
Scott, Sinclair & Co.,	...	343	342	125	18	13	259	15·10	70	5	0	18	14	7	1¾	42	1	9	5	8	9½	3·51	8	11
Do. Do.	Side Lever.	313	312	140	18	15	276	17·63	65	6	0	18	13	7	1¾	52	4	21	0	13	5	4 47	8	11
Do. Do.	Oscillating.	313	312	125	18	13	259	16 55	65	6	0	18	13	7	1¾	47	9	15	11	12	7½	4·33	8	11
Seaward & Capel,	Direct.	469	470	130	44	20	300	12 79	81	5	4½	21	15	5½	2	49	1	19	1½				8	9½

TABLE VI.—PROPORTIONS OF TUBULAR MARINE BOILERS, AS SHEWN BY THE PRACTICE OF THE PRINCIPAL CONSTRUCTING ENGINEERS.

NAMES OF THE ENGINEERS.	Horse Power, Contractor's Estimate.	Horse Power, Admiralty Estimate.	Estimated weight of Boilers and Apparatus.	Estimated weight of Water in Boilers.	Estimated weight of Coal Boxes.	Weight of Coals carried in Boxes.	No. of days' consumption.	Estimated consumption per horse per hour.	Length of Boilers over all.		Breadth of Boilers.		Height of Boilers without Steam Chest.		Height of Boilers with Steam Chest.		Breadth of Fires.		Area of fire-bar surface.	Ratio, per Admiralty horse power.	Length of Tubes.		External diameter of Tubes.	Number of Tubes (iron).	Total area through the Tubes	Total effective heating surface.	Ratio of heating surface, per Admiralty horse power.	Number of Fires.
	No.	No.	Tons.	Tons.	Tons.	Tons.	Days.	Lbs.	ft.	in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.	sq. ft.	Ratio.	ft.	in.	Inches.	No.	sq. ft.	sq. ft.	Ratio.	No.
Boulton & Watt,	480	484	71	31	11	400	15		18	7	19	1	9	7	14	6	2	6			6	6	3					12
Bury, Curtis & Kennedy,	426	427	60	38	16	304	11·1	6	20	2	22	1	12	0	18	2	2	7	191	·448	6	10	3	924	6531	6499	15·22	12
Butterley Company,	420	423	51	48	10	349	12¾	6	20	0	18	6	12	3	15	6	3	5	210	·498	5	9	3	980	6927	5868	13·87	8
Caird & Co.,	500	496	50	50	20	340	13	5	21	0	23	2	12	0	17	6	2	8	240	·484	6	6	3¼	1056	7964	7210	14·54	12
Fairbairn & Co.,	460	463	48	40	15	300	10½	6	16	3	25	0	12	8	15	6	2	10	268	·578	5	7	3	1008	7125	6479	14·00	14
Forrester & Co.,	460	460	69	46	23	360	12¼	6	21	0	23	9	11	6	16	0	2	3	288	·626	6	0	3¼	1134	7632	6484	14·10	16
Fawcett & Preston,	450	444	70	50	19	360	11⅔	6½	22	0	19	6	11	6	16	0	2	3½			6	11	3¼	792	6570	6214	14·00	12
Maudslay, Sons & Field,	450	456	72	41	8	320	10	7	19	0	24·	0	11	8	16	2	2	6			5	0	2½	1900	8595	6218	13·64	12
Miller & Ravenhill,	544	544	78	48	16	250	9	5	23	3	24	6	12	3	16	6	2	9½	240	·441	7	0	3¼			5500	10·11	12
Napier, Robert,	500	508	65	52	20	300	·9	6	19	9	27	0	11	8	16	8	2	6	323	·502	6	2	3	768	5428	6019	11·85	16
Penn & Son,	500	501	70	28	16	340	9.1	7	19	0	19	0	10	4	14	6	1	10½	263	·524	6	6	3	1040	6869	6776	13·52	16
Rennie, J. & G.,	500	501	68	46	15	344	9	7	20	0	24	0	11	0	15	4	2	4	264	·528	6	0	3⅛	984	6388	6421	12·82	16
Rigby, John,	510	515	73	35	17	345	9½	6½	18	6	23	0	14	2	18	0	2	3½	284	·552	6	4	3	1232	7833	8349	16·21	16
Scott, Sinclair & Co.,	342	343	50	40	13	380	15	7	17	10	19	0	12	10	17	6	2	6	195	·568	6	0	3½	880	6931	5656	16·49	12
Seaward & Capel,	470	470	62	27	17	325	12	5½	15	6	23	6	11	0	11	0			287	·612								

TABLE VII.—DIMENSIONS OF CONDENSING STATIONARY ENGINES, BY MESSRS. HICK OF BOLTON.

Horses' Power.	Diameter of Cylinder.	Length of Stroke.		Cylinder Ports.		Air Pump.			Delivery Valve.		Foot Valve.		Crank Neck.		Swivels.		Diameter of Piston Rod.	Main Links.				Back Links.		Diameter of Radius Rod.	Diam. of Air-pump rod.	Diam. of hot-water Pump.	Diam. of cold-water Pump.	Beam Centres.		Crank Pin.		Diameter of Condenser.
				long.	broad.	diam.	stroke.		long.	broad.	long.	broad.	diam.	long.	diam.	long.		long.		wide.	thick.	wide.	thick.					diam.	long.	diam.	long.	
	in.	ft.	in.	in.	in.	in.	ft.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	ft.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
4	12½	2	6	4¾	1⅛	8	1	3	7½	3½	5½	3½	2¼	3¼	1¾	2¼	1⅜			1¾	⅜	1¼	⅜	¾	1¼	2⅜	4	2⅝	2¾	2	3	10
6	14½	3	0	6	1¼	9	1	6	8¼	3½	6¼	3½	2½	4	2	2½	1⅝					1¼	⅜	¾	1⅜	2⅜	4½	2⅞	3	2	3½	11
8	17	3	0	6	1¾	11	1	6	9	3½	7	3½	4½	6	2¼	2½	1⅞	1	6	2	½	1½	⅜	⅞	1⅜	2½	5	3¼	3½	2¼	3¾	13¼
12	19½	4	0	10	1½	12½	2	0	10	3½	9	3½	5½	7	2¾	2¾	2	1	7	2	⅝	1½	⅜	1	1⅜	2½	5½	3½	4	2½	4½	15
15	24	3	0	9½	2½	13½	1	6	10	4	9	4	6	8	2¾	2¾	2½	1	6	2	⅝	1⅝	⅜	1	1½	2¾	6	3¾	4	2¾	4¾	10×20
15	21	4	0	10	1½	13½	2	0	10	4	9	4	6¼	8¼	2¾	2¾	2½	1	7	2	⅝	1⅝	⅜	1	1½	2¾	6	3¾	4	2¾	4¾	16¼
20	27	3	0	11	2½	15	1	6	12	5	12	4	6½	8⅞	3	3	2⅞	1	6	2¼	⅝	2	⅜	1⅛	1¾	2¾	7	4¼	6	3	5½	12×20
20	24½	5	0	10	2	15½	2	6	12	5	12	4	7¼	9	3	3	2⅞	2	0	2	¾	2	⅜	1⅛	1¾	2¾	7	4¼	6	3	5½	18½
25	27	5	0	10	3	17½	2	6	14	5	12	4¼	8	10	3¾	3¾	3	2	6	2½	¾	2¼	⅜	1⅛	2	3¼	8	5	6	3¼	6	19¼
30	32	3	0	14	3	18	1	6	15	5½	12	5	7¾	9¾	3½	3½	3¼	1	6	2½	¾	2¼	7/16	1¼	2	3½	8	5½	6	3½	6	24×25
30	29	6	0	11	3	18	3	0	15	5½	12	5	9	11	3½	3½	3¼	2	6	2½	¾	2¼	7/16	1¼	2	3½	8	5½	6	3½	6	20
40	36	3	6	15	3½	20	1	9	16	6½	15	6	10	12	4	4	3½	1	9	3	¾	2¼	½	1⅜	2¼	4	10	6	8	4	7	22
40	33	6	0	14	3	21	3	0	16	6½	15	6	10	12	4	4	3½	2	6	3	¾	2¼	½	1⅜	2¼	4	10	6	8	4	7	24
50	40	4	0	16	3¾	22	2	0	17	6½	16	6	10	12	4¼	4¼	4	2	0	3½	¾	2¼	9/16	1½	2½	4	11	6½	9	4¾	7	24
60	43	4	0	18	4	24	2	0	21½	6½	18	6	11	14	4½	4½	4	2	0	3½	⅞	2¼	⅝	1½	2½	4½	12	6	8	4	7	24×38¾
60	40	7	0	16	3¾	25	3	6	21	6½	18	6	12½	16	4½	4½	4	3	0	3½	⅞	2¼	⅝	1⅝	2½	4½	12	7	9	5	8	28
70	43	7	0	17	3¾	27	3	6	24	6½	19	6½	12¾	16	4¾	4¾	4¼	3	0	3¾	⅞	2½	½	1⅝	2⅝	5½	12	7½	9	5½	8½	31
80	46	7	0	18	4	29	3	6	24	7½	20	7	13	16	5	5	4½	3	0	4	⅞	2½	9/16	1⅝	3	6	13½	8	10	6	8½	31

TABLE VIII.—DIMENSIONS OF STATIONARY ENGINES CONSTRUCTED BY MR. WILLIAM FAIRBAIRN.

NOMINAL POWER OF THE ENGINE.	100 H. P.	80 H. P.	60 H. P	30 H. P
Diameter of cylinder, ...	54 inches.	46 inches.	39⅞ inches.	30 inches.
Length of the stroke, ...	7 ft. 6 in.	7 ft. 6 in.	7 feet.	6 feet.
Number of strokes per minute, ...	16½	16½	17½	21
Diameter of air-pump, ...	28½ in.	24⅝ in.	24½ in.	10¾ in.
" condenser, ...	30 —			22 —
" cold water pump, ...	16 —	13 —		30 —
" piston rod, ...	5½ —	4½ —	4⅛ —	3 —
" air-pump rod, ...	3¼ —	2½ —	2¾ —	1⅞ —
" crank pin, ...		6¾ × 7½ long.	6¼ —	4¾ —
" crank shaft (for two coupled engines), ...	13½ × 17 long.		13½ —	
Area of steam ports, ...	150 sq. in.	120 sq. in	76 sq. in.	40 sq. in.
" foot and delivery valves, ...	200 —			60 —
" injection valve, ...	33·2 —			10 —
Length of parallel motion main links, centre to centre, ...	4 feet.	Governor, length of arms from point of suspension, 3 feet—making 30·3 revolutions per minute.		Steam worked at 20 to 30 lbs. above the atmosphere, and may be used expansively from ¼ of the stroke.
Section of do. do.	4½ in. × 1½ in.			
Diameter and length of bearings, ...	5 in. × 5¾ in.			
Section of back links of parallel motion, ...	3 in. × ¾ in.			
Diameter and length of bearings, ...	2¾ in. × 3⅝ in.			

TABLE IX.—PROPERTIES OF STEAM, AND MECHANICAL EFFECT OF A GIVEN UNIT AT DIFFERENT PRESSURES.

PRESSURES.		Temperature in degrees Fahr.	Weight of a cubic foot of steam in lbs.	Velocity of exit in feet.	MECHANICAL EFFECT WITHOUT CONDENSATION.				MECHANICAL EFFECT WITH CONDENSATION.			
In Atmospheres.	In lbs. per square inch.				Without expansion.	Expansion at one-half.	Expansion at one-third.	Expansion at one-fourth.	Without expansion.	Expansion at one-half.	Expansion at one-third.	Expansion at one-fourth.
1·00	14·70	212 00	0·0364	0	0	—32·4	—95·2	—170·5	91·3	150·1	178·6	194·6
1·25	18·38	223·88	0·0440	873	21·5	10·1	—32·3	— 87·4	95·9	158·7	190·6	209·9
1·50	22·05	234·32	0·0529	1135	36·4	39·3	10·8	— 30·6	99·3	165 2	199·6	221·1
1·75	25·72	242·78	0·0609	1295	47 4	60·8	42·5	11·1	102·0	170·0	206 2	229·5
2·00	29·40	250·79	0·0688	1407	55·9	77·5	67·0	43·2	104·3	174·2	212·0	236·5
2·25	33·08	257·90	0·0766	1491	62·8	90·9	86·5	68·8	106·2	177·7	216·7	242·4
2·50	36·75	263·93	0·0344	1556	68·4	101·8	102·4	89·6	107·7	180·5	220·5	247·1
2·75	40·42	269·87	0·0921	1608	73·1	111·0	115·8	107·1	109·3	183·2	224·2	251·6
3·00	44·10	275·00	0·0998	1652	71·1	118·8	127·1	121·9	110·5	185·4	227·7	255 2
3·25	47·78	279·86	0·1073	1690	80·7	125·6	137·1	136·7	111·7	187·6	230·0	258·7
3 50	51·45	284·63	0·1148	1722	83·8	131·5	145·6	145·8	112·7	189·4	232·4	261·6
3·75	55·12	288·66	0·1225	1750	86·5	136·8	153·2	155·6	113·7	190·1	234 7	264·4
4·00	58·18	292·91	0·1298	1774	89·0	141·5	160·0	164·5	114·6	192·8	236 9	267·0
4·50	66·15	300·27	0·1445	1816	93·2	149·8	171·5	179·4	116·2	195·6	240·5	271·4
5·00	73·50	307·94	0·1590	1850	96·8	156·5	181·6	192·0	117·7	198·3	244·1	275·6
6·00	88·20	320·00	0·1878	1904	102·5	167·2	196·5	211·4	120·2	202·6	249·7	282·2
7·00	102·90	331·56	0·2159	1945	107·0	175·6	208 4	226 5	122·4	206·4	254·6	288·1
8·00	117·60	340·83	0·2436	1978	110·6	182·4	217 9	238·4	124·3	209·7	258·8	292·1
9·00	132 30	351·32	0 2708	2006	113·7	188·2	225·9	248·5	126·0	212·8	262·7	293·6
10·00	147·00	359·60	0·2977	2029	116·3	193·0	232·5	256·7	127 5	215·3	266·0	301·4
12·50	183·75	377·42	0·3642	2074	121·5	202·5	245·5	273·0	130·7	220·8	272·9	309·5
15·00	220·50	392·90	0·4288	2109	125·7	210·0	255·6	285·4	133·4	225·5	278·9	316·4
17·50	257·25	406·40	0·4924	2136	129·0	216·0	263·6	295·2	135·7	229·5	283·9	322·2
20·00	294 00	418·56	0·5549	2159	131·8	221·0	270·3	305·3	137·8	233·0	288·3	327·2
25·00	367·50	429·34	0·6775	2196	136 3	229·1	281·0	316·2	141·2	238·9	295·7	335·8
30·00	441·00	457·16	0·7970	2226	140 0	235·6	289·5	326·4	144·2	244·0	302·0	343·1

TABLE X.—TEMPERATURE AND VOLUME OF THE STEAM GENERATED UNDER DIFFERENT PRESSURES, COMPARED TO THE VOLUME OF THE WATER THAT HAS PRODUCED IT.

PRESSURES.		Corresponding temperature, by Fahrenheit's thermometer.	Relative volume, or volume of the steam compared to the volume of the water that has produced it.	PRESSURES.		Corresponding temperature, by Fahrenheit's thermometer.	Relative volume, or volume of the steam compared to the volume of the water that has produced it.	PRESSURES.		Corresponding temperature, by Fahrenheit's thermometer.	Relative volume, or volume of the steam compared to the volume of the water that has produced it.	PRESSURES.		Corresponding temperature, by Fahrenheit's thermometer.	Relative volume, or volume of the steam compared to the volume of the water that has produced it.
In lbs. per sq. inch.	In inches of mercury.			In lbs. per sq. inch.	In inches of mercury.			In lbs. per sq. inch.	In inches of mercury.			In lbs. per sq. inch.	In inches of mercury.		
1	2·04	102·9	20954	29	59·16	249·2	909	57	116·28	290·7	490	84	171·38	316·9	344
2	4·08	126·1	10907	30	61·21	251·2	882	58	118·32	291·9	482	85	173·42	317·8	340
3	6·12	141·0	7455	31	63·24	253·1	855	59	120·36	293·0	474	86	175·46	318·6	337
4	8·16	152·3	5695	32	65·28	255·0	831	60	122·40	294·1	467	87	177·50	319·4	333
5	10·12	161·4	4624	33	67·32	256·8	808	61	124·44	294·9	460	88	179·54	320·3	330
6	12·24	169·2	3901	34	69·36	258·6	786	62	126·48	295·9	453	89	181·58	321·1	326
7	14·28	176·0	3380	35	71·40	260·3	765	63	128·52	297·0	447	90	183·62	321·9	323
8	16·32	182·0	2985	36	73·44	262·0	746	64	130·56	298·1	440	91	185·66	322·7	320
9	18·36	187·4	2676	37	75·48	263·7	727	65	132·60	299·1	434	92	187·70	323·5	317
10	20·40	192·4	2427	38	77·52	265·3	710	66	134·64	300·1	428	93	189·74	324·3	313
11	22·44	197·0	2222	39	79·56	266·9	693	67	136·68	301·2	422	94	191·78	325·0	310
12	24·48	201·3	2050	40	81·60	268·4	677	68	138·72	302·2	417	95	193·82	325·8	307
13	26·52	205·3	1903	41	83·64	269·9	662	69	140·76	303·2	411	96	195·86	326·6	305
14	28·56	209·0	1777	42	85·68	271·4	647	70	142·80	304·2	406	97	197·90	327·3	302
15	30 60	213·0	1669	43	87·72	272·9	634	71	144·84	305·1	401	98	199·92	328·1	299
16	32·64	216·4	1572	44	89·76	274·3	620	72	146·88	306·1	396	99	201·96	328 8	296
17	34·68	219·6	1487	45	91·80	275·7	608	73	148·92	307·1	391	100	204·01	329·6	293
18	36·72	222·6	1410	46	93·84	277·1	596	74	150·96	308·0	386	110	224·40	339·2	271
19	38·76	225·6	1342	47	95·88	278·4	584	75	153·02	308·9	381	120	244·82	345·8	251
20	40·80	228·3	1280	48	97·92	279·7	573	76	155·06	309·9	377	130	265·23	352·1	233
21	42·24	231 0	1224	49	99·60	281·0	562	77	157·10	310·8	372	140	285·61	357·9	218
22	44·88	233·6	1172	50	102·00	282·3	552	78	159 14	311·7	368	150	306·03	363·4	205
23	46·92	236·1	1125	51	104·04	283·6	542	79	161·18	312·6	364	160	326·42	368·7	193
24	48·96	238·4	1082	52	106·08	284·8	532	80	163·22	313·5	359	170	346·80	373·6	183
25	51 00	240·7	1042	53	108·12	286·0	523	81	165·26	314·3	355	180	367·25	378·4	174
26	53·04	243·0	1005	54	110·16	287·2	514	82	167·30	315·2	351	190	387·61	382·9	166
27	55·08	245·1	971	55	112·20	288·4	506	83	169·34	316·1	348	200	408·04	387·3	158
28	57·12	247·2	939	56	114·24	289·6	498								

TABLE XI.—MECHANICAL PROPERTIES OF THE COMMON METALS AND PRINCIPAL WOODS.

IN THE FOLLOWING TABLE, THE INITIAL LETTERS AFFIXED TO THE NUMBERS INDICATE THE FOLLOWING AUTHORITIES: NAMELY,

B....BARLOW.	Br...BELIDOR.	H....HODGKINSON.	M...MUSCHENBROCK.	R....RONDELET.	Td...TELFORD.
Bl...BRUNEL.	C.....CAUCH.	K....KIRWAN	Mt..MUSHET.	Re...RENNIE.	Tr...TREDGOLD.
Bn...BEVAN.	F.....FAIRBAIRN.	L....LAME'.	P....PASLEY.	T.....THOMSON.	

NAMES OF MATERIALS.	Specific gravity.	Weight of 1 cubic foot in lbs.	Tenacity per sq. inch in lbs.	Crushing force per square inch in lbs.	Modulus of elasticity in lbs.	Modulus of rupture in lbs.	[illegible]
Brass (cast)	8·399	525·00	17,968 Re	10,304 Re	8,930,000		[illegible]
Copper (cast)...	8·607	537·93	19,072				
ditto (sheet)	8·785	549·06	——				
ditto (wire-drawn)	8·878	560·00	61,228				
ditto (in bolts)	——	——	48,000				
Iron (English wrought)	7·700	481·20	25⅓ tons L		24,920,000		
ditto (in bars)	7·760 7·800	475·50 487·00	—— 25½ tons L				
ditto (hammered)			30 tons Bl				
ditto (Russian) in bars			27 tons L				
ditto (Swedish) in bars...			32 tons R				
ditto (English) in wire, 10th inch diameter, ...			36 to 43 tons Td				
ditto (Russian) in wire, 1-20th to 1-30th inch diameter			60 to 91 tons L				
ditto (rolled in sheets and cut lengthwise)			14 tons Mt				
ditto (cut crossways)			18 tons Mt				
ditto in chains, oval links, 6 inches clear, iron ½ inch diam.			21½ tons Br				
ditto (Brunton's) with stay cross link			25 tons B				
Cast-Iron (Old Park)					[illegible] T	43,240 T	
ditto (Adelphi)					[illegible] T	45,360 T	
ditto (Alfreton)					[illegible] T	44,046 T	
ditto (scrap)					[illegible] T	45,828 T	
ditto (Carron, No. 2) hot blast	7·046 H	440·37	13,505 H	108,540 H	[illegible] H	37,503 H	[illegible]
ditto (do. do.) cold blast	7·066 H	441·62	16,683 H	106,375 H	[illegible] H	38,556 H	[illegible]
ditto (do. No. 3)	7·094 F	443·37	14,200 H	115,442 H	[illegible] F	35,980 F	[illegible]
ditto (do. do.) hot blast	7·056 F	441·00	17,755 H	133,440 H	[illegible] F	42,120 F	7·515 : 1
ditto (Devon, No. 3) cold blast	7·295 H	455·93			[illegible] H	36,288 H	
ditto (do. do.) hot blast	7·229 H	451·81	21,907 H	145,435 H	[illegible] H	43,497 H	
ditto (Buffrey, No. 1) cold blast	7·079 H	442·43	17,466 H	93,366 H	[illegible] H	37,503 H	[illegible]
ditto (do. do.) hot blast	6·998 H	437·37	13,434 H	86,397 H	[illegible] H	35,316 H	[illegible]
ditto (Coed-Talon, No. 2) cold blast	6·955 F	434·06	18,855 H	81,770 H	[illegible] F	33,104 F	[illegible]
ditto (do. do.) hot blast	6·968 F	435·50	16,676 H	82,739 H	[illegible] F	33,145 F	[illegible]
ditto (do. No. 3) cold blast	7·194 F	449·62			[illegible] F	43,541 F	
ditto (do. do.) hot blast	6·970 F	435·62			[illegible] F	40,159 F	
ditto (Milton, No. 1) hot blast	6·976 F	436·00			[illegible] F	28,552 F	
ditto (Muirkirk, No. 1) cold blast	7·113 F	444·56			[illegible] F	35,923 F	
ditto (do. do.) hot blast	6·953 F	434·56			[illegible] F	33,356 F	
ditto (Elsicar, No. 1) cold blast	7·030 F	439·37			[illegible] F	34,862 F	
Lead (English cast)	11·446 M	717·45	1,824 Re		720,000 Tr		
ditto (milled sheet)	11·407 T	712·93	3,328 Tr				
ditto (wire drawn)	11·317 T	705·12	2,581 M				
Silver (standard)	10·312 T	644·50	40,902 M				
Mercury (at 32°)	13·619	851·18					
ditto (at 60°)	13·580	848·75					
Steel (soft)	7·780	486·25	120,000				
ditto (razor tempered)	7·840	490·00	150,000		29,000,000		
Tin (cast)	7·291 Tr	455·68	5,322 M		4,608,000 Tr		
Zinc (cast)	7·028	439·25			13,680,000 Tr		
ditto (rolled)	7·215	450·9					
PRINCIPAL WOODS.							
Acacia (English)	0·71 B	44·37	16,000 Bn		1,152,000 B	11,202 B	
Beech New	0·854	53·37	15,784 B	7,733 H	1,353,600 B	[illegible] B	0·55 : 1
Dry	0·690	43·12	17,850 B	9,363 H			
Birch Common	0·792 B	49·50	15,000	6,402 H	1,562,400 B	[illegible] B	0·45 : 1
American	0·648 B	40·50		11,663 H	1,257,600 B	9,624 B	
Deal Christiana middle	0·698 B	43·62	12,400		1,672,000 B	9,864 B	
Memel middle...	0·590 B	36·87			1,535,200 B	[illegible] B	
Norway spruce	0·340	21·25	17,600				
English	0·470	29·37	7,000				
Elm (seasoned)	0·588 C	36·75	13,489 M	10,331 H	699,840 B	6,078 B	0·79 : 1
Fir New England	0·553 B	34·56			2,191,200 B	6,612 B	
Riga	0·753 B	47·06	12,000	6,000			[illegible]
Larch (seasoned)	0·522	32·62	10,220 B	5,568 H	1,052,800 B	6,894 B	[illegible]
Lignum-vitæ	1·220	76·25	11,800 M				
Mahogany (Spanish)	0·800	50·00	16,500	8,198 H			[illegible]
Oak English	0·934 B	58·37	17,300	4,684 wet 9,504 dry H	1,451,200 B	10,032 B	[illegible] [illegible]
Canadian	0·872 B	54·50	10,253	4,231 wet [illegible] dry H	2,148,800 B	10,596 B	[illegible] [illegible]
Dantzic	0·756 B	47·24	12,780		1,191,200 B	8,742 B	
Pine Pitch	0·660 B	41·25	7,818 M		1,225,600 B	9,792 B	
Red	0·657 B	41·06		5,375 H	1,840,000 B	8,946 B	
Yellow	0·461 C	28·81		5,445 H	1,600,000 Tr		
Plane-tree	0·64	40·00	11,700 Bn				
Poplar	0·383 M	23·93	7,200 Bn	3,107 wet 5,124 dry H			[illegible] [illegible]
Teak (dry)	0·657 C	41·06	15,000 B	12,101 H	2,414,400 B	14,772 B	0·81 : 1
Willow (dry)	0·390	24·37	14,000 Bn				
Yew (Spanish)	0·807	50·43	8,000 Bn				

ENGINEER AND MACHINIST'S ASSISTANT.

PART SECOND.

ON MILL GEERING.

Geering is the general term employed to denote a combination of mechanical organs, interposed between the prime mover and the working parts of machinery. Frequently, however, the signification is restricted to the series of toothed wheels by which the motion is conducted from one revolving axis to another, independently of the shafts and bearings by which they are supported. Two toothed wheels are also said to *geer* when they have their teeth engaged together, and to be *out of geer* when separate and consequently out of action.

A rigid definition of the term is therefore not to be expected, and is indeed of little moment in a practical point of view. We employ it here in its extended and general acceptation, as a name comprehending all those parts destined to operate in the transmission of the motive power to the working parts of the mechanism; but shall have frequent occasion to use it in its more limited signification, as a convenient way of expressing the mode of action of one toothed wheel upon another. The context will in all cases sufficiently determine the particular sense in which the word is to be taken, without danger of confusion; and nothing further is required for the purpose contemplated.

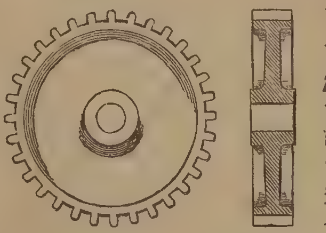
SECTION I.—VARIETIES OF TOOTHED GEER.

In the transfer of motion from one axis to another in a system of mechanism, wheels are the most important organs. Of these there are two principal varieties with several modifications, distinguished by peculiarities of form, construction, and adaptation, which it will be necessary to describe as we proceed.

It may, however, be of importance to premise in the meantime, that whatever be the form of a pair of wheels, their action is strictly analogous to that of two cylindrical or conical surfaces rolling upon one another. The ridges, or *teeth*, placed upon the peripheries are simply a convenient and certain method of maintaining the contact. The teeth being disposed at equal distances on the rims of the two wheels, when the driving wheel is turned its teeth enter successively the spaces between those of the wheel with which it geers, and thereby prevents all slipping of the surfaces, notwithstanding slight errors of form. Could the simple adhesion of the surfaces be sufficiently augmented, without other inconveniencies, to insure the required condition of uniform action—and in some instances this is successfully accomplished—then toothed wheels might be dispensed with, and the construction of geering thereby greatly simplified. But in general practice the contrivance of teeth is by far the most convenient and effectual method of attaining the object, and is therefore of corresponding importance in constructive mechanics.

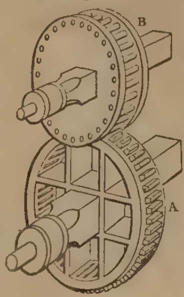
When motion is to be transferred from one axis to another which is parallel to it, the peripheries of the wheels act upon each other tangentially, and the teeth are disposed round their cylindrical surfaces in the direction of radii from their centres. Wheels of this sort take the general name of *spur geer*, of which there are several modified forms and modes of construction, depending chiefly on the purposes to which they are applied, and the kind of material of which they are composed. In the ordinary cast-iron wheels, the teeth are made in one piece with the *rim;* and commonly the wheel consists of one entire piece, though sometimes the wheel is cast in segments when of very large size, and especially if intended for shipment, for convenience of packing and transport. The best modern forms of wheels constructed in these two ways will be minutely described when we come to treat of the construction of wheels, and therefore in the meantime we simply remark, that cast-iron wheels of all sizes above, and frequently below 18 inches diameter, are made with arms in the manner shown in several examples of geering given in the Plates; but in very small sized wheels the arms are commonly replaced by a continuous plate or *web*, which unites the toothed rim with the eye-boss at the centre, as represented by the annexed cuts. Wheels of this kind are technically denominated *plate-wheels*, to distinguish them from those of similar size which have arms. The wheels of lathes, and indeed the toothed geer of the smaller class of machine-tools of textile machinery, and of the more highly finished class of small machines generally, are commonly made in this manner, the wheel in those cases being usually face-turned and polished. When the pitch is very fine, the wheel is commonly cast from a blank pattern; the circumference is then turned, and the teeth cut in the solid rim by the wheel-cutting machine. In this manner greater accuracy is attainable, but the teeth are not usually equal in strength and durability to those of a cast-wheel.

Fig. 1.



In wooden wheels, which were in general use for all species of millwork till towards the end of last century, and of which we have still some examples, the teeth are formed of separate pieces, and fixed into equidistant mortises pierced through the rim of the wheel. A of fig. 2, represents the construction of a wheel of this sort, which was only recently replaced by a cast-iron wheel. The rim was formed of segments of hard-wood firmly bound together by iron straps and bolts, and was connected to the shaft by a wooden framing, consisting of bars set at right angles. The square opening, thus left at the centre to receive the shaft, which was also of wood, and square at that part, was purposely made larger than the section of the shaft, to admit of adjustment and fixing by wedges. In other examples the arms are mortised into the shaft; but such a mode of fixing is very injudicious, as it not only weakens the shaft, but also renders adjustment and any subsequent alteration in the arrangement extremely difficult.

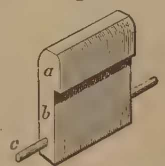
Fig. 2.



Wheels of this kind are still in very common use on the continent, especially in localities where iron wheels cannot be easily obtained, and possibly also on account of their being less costly. In this country they are technically known as *cog-wheels*, and the teeth being formed of separate pieces take the name of *cogs*. These are made of some well-seasoned hard-wood, as mountain-beech, plane-tree, hornbeam, hickory, and the like, with the grain disposed in the direction of their length, which being the radial direction, is the most favourable to transverse strength.

A modification of this construction of toothed wheels is still very commonly employed in millwork under the name of *mortise-wheels*. In these, the body of the wheel is of cast-iron, and the teeth of wood, fixed into mortises made in the rim, as in the old cog-wheel. The individual teeth, or cogs, are often formed in the manner represented in fig. 3; the part *a* projects without the rim, and is strictly the tooth; the tenon *b* is made to fit its mortise in the rim of the wheel very tightly, and is left sufficiently long to project on the inside, so that being driven into the mortise up to the shoulder of *a* it is secured in its place by an iron pin *c*, which passes either closely under, or through the rim of the wheel.

Fig. 3.



There is, however, another mode of fixing the cogs, which is at present, perhaps, more commonly practised than that described, though less simple, and possessing no real superiority. This mode will be understood by reference to Plate VII. from which it will be observed, that the cogs instead of being made fast in their places by pins, are retained in the mortises of the rim by dovetail keys v, v, driven between every contiguous pair inside of the rim. The tenons, when the cogs are intended to be fitted in this manner, are made with an expanding dovetail at their extremities, to receive one side of the key, which being driven tightly into the V-shaped space, thus formed between the ends of each pair inside of the rim of the wheel, retains the cogs in their places. The keys, which are made of well-seasoned hardwood, usually of the same kind as that used for the cogs, are made long enough to project some way beyond the side of the rim, to allow of their being driven in more tightly, should the cogs in the course of working become loose. The ends of the tenons are also very commonly made *flush* with the surface of the keys, but sometimes they are made to project beyond, with a view to strength.

This mode of fitting by keys admits of much neatness of finish, but is less convenient than that before described, at those points where the arms of the wheel meet the rim. At those points it is necessary to have recourse to fixing by simple wedges, which rarely admit of the requisite degree of security against loosening of the cogs in working; in other respects it is perfectly effectual, and is still, by some, reckoned an improvement on the more simple mode of fixing the cogs individually by iron pins, which was formerly the universal practice.

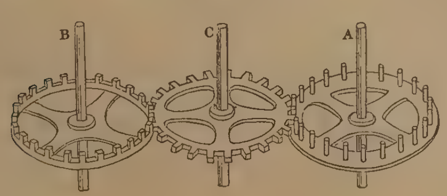
The mortise-wheel is at present preferred, where a high velocity with smoothness of motion is required. It is usually made to work into a wheel with cast-iron teeth, a pair of this sort being found to work together with less vibration, and consequently less noise, and that the teeth abrade one another less, and therefore endure longer, than when both wheels of the pair have iron teeth. For these reasons it is very common, in good millwork, to make one wheel of every large-sized pair with wooden cogs, especially when the speed is high, and subject to variations of velocity. This practice is indicated graphically by the bevil pair of wheels depicted in Plate VII.; and is very fully illustrated by the flour-mill geering shown in Plates XCIX.—CVI. But it may be here remarked, that in cases where a very large wheel is required to geer with a smaller one, the former is commonly made the mortise-wheel, except when on account of some peculiarity it is necessary to construct it of segmental pieces. We have examples of this arrangement in many of those cases in which, for the purpose of bringing up the speed of an engine directly, that is, without intermediate geering, the main-shaft is driven from the circumference of the fly-wheel. The fly-wheel is then constructed with toothed segments fixed round its periphery, the teeth being disposed as in any other spur-wheel, and is made to work into a mortise-wheel of the requisite size placed on the main shaft. On the contrary, the large wheel in a thrashing-mill has very commonly wooden teeth, which geer with the iron teeth of the smaller wheel placed upon the main-drum axle.

By this arrangement the full advantage contemplated is attained more effectually than it would be by the inverse order, in consequence of the difference in form between the teeth of a large and small wheel of the same *pitch*, as will appear when treating of the form of the teeth of wheels; but in the case of the power of a water-wheel, or of a steam-engine, being taken off at the circumference, the large segmental cast-iron wheel becomes of more easy construction, and is therefore more economical as respects first cost, and also as respects durability, than a mortise-wheel of equal dimensions. There are, besides, practical objections to the mortise-wheel, as the weakening of the rim by cutting out the tenon mortises, and of diminishing its weight, which is of importance in the case of the wheel fulfilling the functions of a fly-wheel. Were it attempted to obviate these objections by depth of metal, another evil would arise from the great length of tenon which it would be necessary to give to the cogs. An increase of breadth would in like manner be accompanied with a difficulty as respects the fixing of the cogs in their places by the keys or pins.

It may be here well to observe, that when two wheels geer together, the one which communicates the motion to the other is called the *driver* or *leader;* and the wheel impelled is the *follower.* If the two wheels be of very different sizes, the smaller one very commonly takes the name of *pinion,* which may be regarded as the diminutive of *toothed-wheel,* and if more convenient, the term *wheel* may be employed without impropriety to designate a pinion. For the sake of further distinction the teeth of pinions are often termed *leaves,* especially by clock-makers. In geering of the ordinary kinds, the pinions are commonly of the same form of construction as the wheels with which they act; but in the old wooden machinery the pinion is commonly formed by inserting the extremities of a number of wooden cylinders into equidistant holes, in two parallel discs (technically called *heads* and *cheeks*) upon a square shaft as represented by B of fig. 2. Pinions of this sort are now not more common than cog-wheels, with which they have in a great measure disappeared from modern practice, except in a few peculiar cases, such as powder-mill geering, and the like. It is also found on a miniature scale in wooden clocks of Dutch construction. A pinion of this description is denominated a *trundle, lantern,* or *wallower,* and its cylindrical teeth are termed *staves,* sometimes *rounds* and *rondles,* very commonly written and pronounced *rungs.* The construction admits of great strength, and with modern workmanship, and the substitution of iron for wood, is susceptible of being employed with advantage, especially when the pinion is to be *driven* by a wheel of very large size; for under these circumstances, as we shall hereafter have occasion to remark more fully, the circular section of its staves gives it a smoothness of motion scarcely attainable with a pinion of the ordinary form.

Professor Willis observes in his Principles of Mechanism, in reference to the trundle, that the system is of great antiquity; for in early machinery the toothed wheels were often cut out of thin metal plates, which rendered it impossible to make a pair thus formed to work together, as is done in modern practice, in which the wheels have considerable breadth on their toothed surfaces, for the slightest deviation of one of the wheels from the plane of rotation of the pair, would allow the teeth to lose hold of each other sidewise. To obviate this, one of the wheels of a pair was always made either in the lantern form, as just described, or more commonly with pins inserted at one end only, into a disc as represented at A in fig. 4. A modification of this method also in use, was to form the teeth on the edge of a hoop, as shown at B. By these arrangements, the thin wheel C was enabled to retain its hold of the teeth of the wheel with which it geered, notwithstanding a little deviation from the plane of rotation, arising either from incorrectness of workmanship, vibration of the disc, or end-play of the axis.

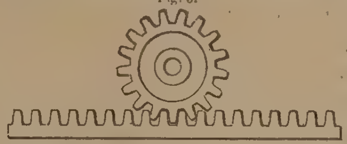
Fig. 4.



The form of wheel represented at B is still in common use in watch and clock-work under the names of *crown* and *contrate-wheel.* Its application is, however, different. Instead of being employed to connect two parallel axes, it is only used in cases where the axes meet at right angles; but its teeth are cut on the edge in the same manner—making allowance for greater accuracy of work, and correctness of adaptation—as the wheel B.

This arrangement of wheel and pinion may be regarded as strictly analogous to the case of a wheel working into a *rack,* (as shown in fig. 5,) the curvature of that portion of the crown-wheel immediately in contact with its pinion, being so small that it may be neglected. The case of the pinion and rack may again be referred to that of a small wheel geering with one of very large size, strictly of infinite diameter. This will come particularly under consideration in speaking of the proper forms of the teeth of wheels.

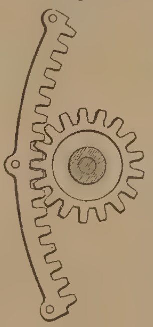
Fig. 5.



If the rack be curved upon the pinion it becomes an *internal* or *annular* wheel, that is, a wheel having the teeth inside of the rim as represented by fig. 6. In this the axes of the wheel and pinion are parallel, and moreover, revolve in the same direction. The arms of the internal wheel are necessarily situated behind the rim to prevent their interference with the pinion, and the latter must overhang its bearing, that is, be fixed on the extremity of its shaft, to avoid interference with the arms of the wheel.

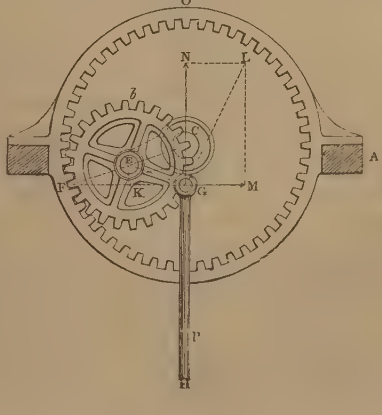
Wheels of this kind are very commonly employed to take off the motive power of large water-wheels. When so applied, the internal wheel is commonly cast in segments, and bolted to the *shrouding,* and the pinion is situated within the loaded arc of the water-wheel, in order to take off the strain from the wheel. Sometimes, indeed, the pinion is awkwardly placed in the contrary position; but unless the circumstances of the case be such as to render an arrangement of that kind indispensably necessary, it is never adopted by an engineer who has any regard to his reputation.

Fig. 6.



When the pinion is made half the diameter, that is, equal in diameter to the radius of the annular wheel, or circular rack, the arrangement admits, in small steam-engines, of a very elegant application, known as White's parallel motion.

Fig. 7.



In this the annular wheel is fixed at AA, and the pinion is attached upon a crank-arm CE which has its centre of motion at C, and carries the centre E. The rod P being attached at G, the circumference of revolution of the pinion is thereby made to describe the right line PN, coinciding with a diameter of the annular wheel, which is therefore equal to the length of the stroke of the engine.

Internal wheels are more difficult to execute as respects the proper form of their teeth, than external spur-wheels; but as we shall afterwards have occasion to show very fully, the action of their teeth with those of the pinion is smoother and less liable to unequal wear.

Similar combinations to that of the crown wheel and pinion were early introduced into mechanism. In examples of this construction, which are still to be found along with cog-wheels and trundles, the cogs are simply disposed on the rim of the wheel, from which they project in lines parallel to the axis, so as to geer with those of an ordinary cog-wheel, having the cogs disposed round the circumference, or with a trundle when one of the axis revolves much quicker than the other. This is exemplified in the hand mill still to be found in some parts of Germany and the north of Europe, as depicted in the annexed figure, in which we have the *face-wheel* W, upon the crank axle, geering with the trundle T upon the millstone axle, which forms an angle of 90° with the former. The same arrangement is still, indeed, common among millwrights in some localities of our own country, where cast-iron is less plentiful, and in consequence less appreciated, than it is in the manufacturing districts. In several parts of Scotland the combination shown in the annexed cut, as a first pair in the geering of a thrashing-mill, is still common; and were it not that the teeth are often ill formed, would work very advantageously. From the mode of action of a face-wheel—which may be regarded simply as a crown-wheel of a larger size—it is easy to perceive, that the contact between the teeth of the pair ought to be at points only, and that the form of the teeth must be adapted to that condition. Were the cross section of the teeth rectangular, it is obvious that the motion would be irregular; for no sooner would a pair of geering teeth pass the plane cutting the two centres, than the relative diameters of the wheels would be changed, in consequence of the change in the position of contact of the teeth, which for the time act together. To avoid this irregularity, the teeth are formed of a curvilinear section, the amount of curvature depending, of course, on the relative diameters of the two wheels. This is a subject, however, to which we shall have occasion to return, and simply refer to it here as a feature by which wheels of this sort are distinguished.

Fig. 8.

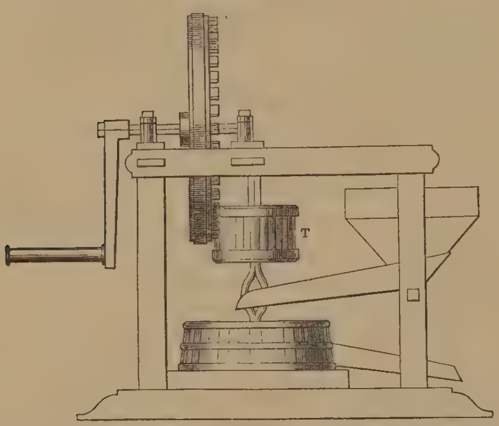
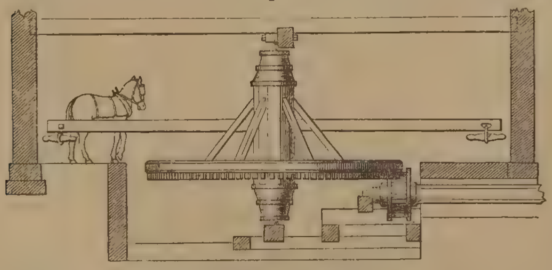
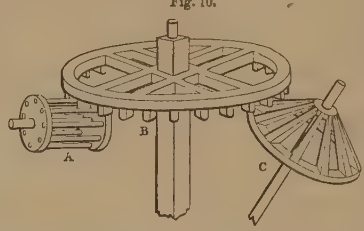


Fig. 9.



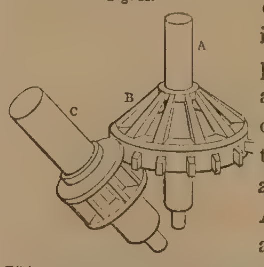
For the purpose of changing the direction of motion through other than right angles, similar modes were adopted by the early mechanicians, all having for their object, to present the teeth to each other at the *plane of centres*, that is, the plane cutting the axes, in the same relative position as in spur geer. Thus, the following arrangement, fig. 10, copied by Professor Willis from De la Hire's Mechanics, (1695,) furnishes a very good illustration of the general character of the contrivances made use of to conduct motion angularly. In showing how the direction of motion may be changed by toothed-wheels, De la Hire observes with reference to this combination, that "the face-wheel B with its trundle A are competent to conduct the motion into a channel at right angles to its primary one." With reference to C, he remarks: "If a lantern C be constructed with its staves inclined to the axis at any given angle, then will the horizontal motion of the power be changed into a motion inclined to it at any angle we please, provided only that the staves of the lantern be so arranged, that they come successively into the horizontal position at the moment of meeting the teeth of the wheel B, in order that they may apply themselves to the teeth in the same manner as if the lantern was like the other."

Fig. 10.



It is rather singular, as Professor Willis observes, that upon the authority of this conical trundle, the invention of bevil geer should have been attributed to De la Hire, when it is plain, that the principle is exactly reversed in his arrangement—the apex of the cone being turned in the wrong direction. It is also known, that the problem of changing the direction of motion through other than right angles had been solved long before the time of De la Hire. Fig. 11 is an example of one of the contrivances in use: it shows part of a complex piece of millwork, extracted from a work published in 1580.[*] The object of the mechanism in question was to conduct the motion of a water-wheel to a series of three Archimedian screws, placed one above the other. A face-wheel

Fig. 11.



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[*] Le Diverse et Artificiose Machine del Capitano A. Ramelli. Paris, 1580.

carried upon the water-wheel shaft, geered with a trundle at the lower extremity of a vertical shaft, which extended to the top of the building, and of which A is a portion. Three conical wheels, similar to B, were placed, one opposite the lower end of each screw, as C, which it turned by geering with a square staved trundle, as shown in the figure.†

These rude modes of accomplishing the object have been almost entirely superseded by the invention of *bevil-geer*, which admits of an almost endless variety of angular position, and consequently of application; and moreover, combines great elegance with almost any required degree of accuracy.

In spur geer, we observe that the principle depends upon two cylindrical surfaces being made to act on each other tangentially; in bevil geer the cylinders are replaced by thin frusta of cones, which have their smooth surfaces exchanged for a regular series of equidistant teeth directed to the apex of the cone; so that a right line passing through the apex, if brought into contact with any part of the side, or top of a tooth, shall touch it throughout its whole breadth; consequently in any pair adapted to work together, the apices of their cones must meet in the same point, and thus the contact of one tooth with another will take place along their sides. This is well illustrated by the pair depicted in fig. 1 of Plate VI. in which the apices of the cones meet at O, to which, it will be observed, all the lines defining the teeth converge. Plate VII. affords another illustration of the same principle; but in this the teeth are formed of wood on the principle of the mortise-wheel already described, and the difference in size between the wheel and pinion is also much greater than it is in the example given in Plate VI. For this reason the wheel approaches the appearance of the old face-wheel, and the pinion assumes somewhat the form of the lantern; but the mode of action is entirely different. In the face-wheel and trundle, the contact of the teeth is between two convex surfaces, and consequently at a point only, whereas the contact is maintained in a bevil pair, as already remarked, along the whole line of breadth of the teeth.

Plate V. exemplifies another variety. In this it will be observed, that the point D, at which the apices of the cones meet from the point F of the wheel, and the point E of the pinion (not drawn but simply indicated), are equal; in other words, that the line DF=the line DE; and further, that these lines are respectively equal, and parallel to the lines BE and BF. These four lines, therefore, enclose a square, of which the line BD is the diagonal, and this being the case, it follows that the wheel and pinion are of equal size,—a circumstance which is expressed by the name *meter-wheels*.

In these examples, motion is led off at an angle of 90°; but as already remarked, bevil geer may be employed to change the direction of the motion to any angle required. This will readily appear from the conditions expressed in the following diagrams, in which, for the sake of simplicity, we suppose the cones continued till their apices meet.

Fig. 12.

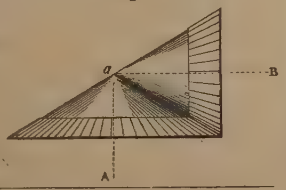


Fig. 12. In this we have an illustration of the principle of those cases already described. Here *a* is the point at which the apices meet, and supposing the motion to be conveyed in the axis A*a*, it is transferred to the axis *a*B in a right angle to its first direction.

Fig. 13.

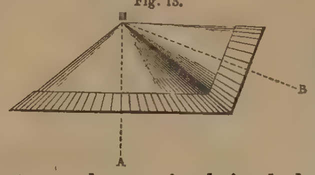


Fig. 13. In this let the letters have the same references as before; it is then plain, that the change of direction is no longer a right angle; for the angle included between the axes is obviously less than 90°, and might evidently be made still less by diminishing the angles which the slanting sides of the cones make with their respective axes. Wheels of this kind—rather wheels of which the angle included by the axes is *less* or *greater* than a right angle, are usually distinguished by the name of *conical-wheels*.

Fig. 14.

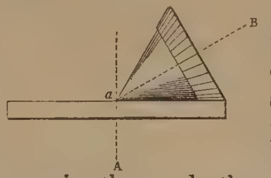


Fig. 14. In the condition expressed by this figure, the apices of the two cones meet in a point, so that one of them becomes a plane surface; its teeth therefore become radial as referred to its apex; in other words, they diverge in the manner of radii, drawn from the centre *a*. The change in the direction of the motion is in this case greater than 90°, and might be further increased, if consistent with the relative velocity of the pair, by increasing the angle of the cone.

Fig. 15.

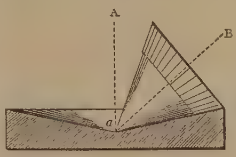
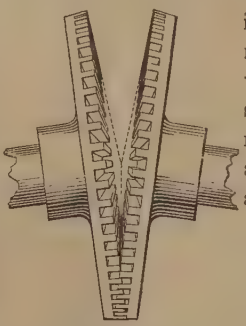


Fig. 15. In this we have an expression of a mode of changing the angle of direction differing from those shown in the preceding figures, inasmuch as one of the cones has become hollow, and the teeth are disposed upon its interior surface, thereby forming an internal bevil-wheel analogous to the internal spur geer previously noticed; wheels are, however, very rarely made in this manner.

Fig. 16.



When the direction approaches a right line, the cones may be made to roll upon each other as in fig. 16. A pair working in this manner is under particularly favourable circumstances as respects strength, and when well and accurately made, the motion is free of all perceptible irregularity. The arrangement admits also of much convenience, as a wheel may often be selected from the pattern room which will "work into itself" sufficiently well, and give the angle required although made, as one of a pair, to transmit a very different angular velocity, and through a very different angle of direction.

From the conditions above enumerated, it will hereafter appear, that it is necessary to include the condition of the relative velocities of the geering pairs—of which in these illustrations we have taken no note. It may, however, be remarked, that the form of the cones to be employed in a bevil pair, must depend upon the two circumstances of velocity and direction, and in practice is not to be deduced from either exclusively

In cases where the axes do not meet in a point as hitherto

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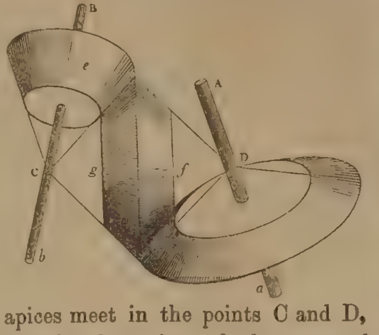
† Willis's Principles of Mechanism, p. 49.

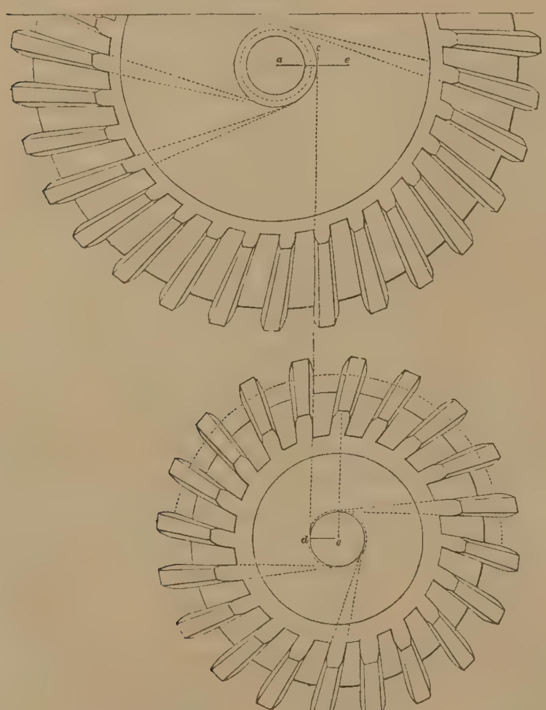
considered, the geering becomes somewhat more complicated either in form or number of parts. The direct mode of arrangement in this case is to employ an intermediate double bevil wheel, as in the annexed fig. 17. Here the two axes A*a* and B*b* do not meet at any point, but are connected by the double bevils *f* and *g*, whose apices meet in the points C and D, and therefore meet severally the apices of the cones of which A and B are the axes, in these points. By the interposition of the bevils *f* and *g*, the condition is therefore fulfilled; for it is evident, that the rotation of the axis A*a* being transferred to the axis CD, will thence pass to the axis B*b* as required.

Fig. 17.



To avoid the multiplication of parts, the object is more commonly attained by a modification in the direction of the teeth, by which they are made to come into contact in an oblique position, answering to the obliquity of the position of the cones upon which they are formed. This artifice is often practised when it is necessary to cross the two axes,—which it must be observed are not theoretical lines but actual shafts of diameters proportionate to the power they are required to transmit. This arrangement is represented by fig. 18 annexed, and illustrates that variety of toothed geer, known as *skew-wheels;* but the mathematical conditions of the form, and disposition of the teeth, must be reserved for subsequent consideration. Wheels of this character, we may however observe, are commonly avoided in practice, unless when a very obvious advantage is to be obtained by their use; as on account of the oblique form of the teeth they are not only of more difficult construction, and therefore rarely possess the same degree of accuracy which we find in bevils of the ordinary sort, but the strain upon the shafts is likewise oblique, and therefore more severe upon the bearings. The whole geering is thereby rendered less durable; it works with more sliding action, endlong pressure, and consequently more friction than the common forms of bevil geer.

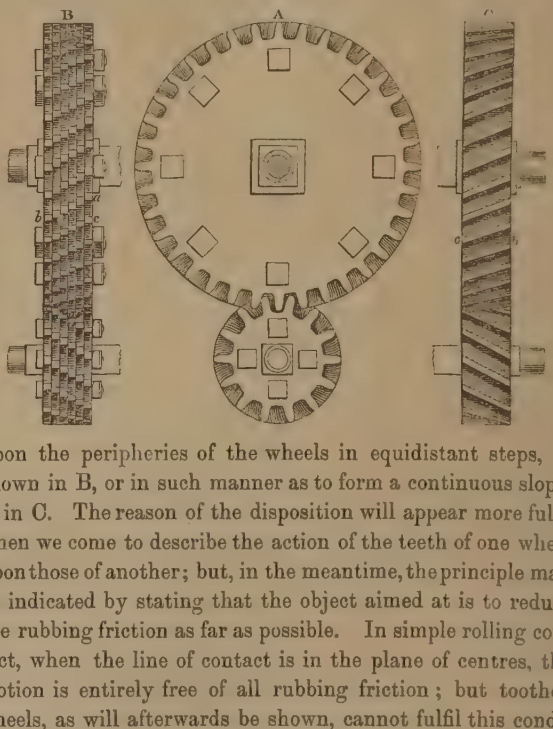
Fig.18.



The purpose of skew-bevils was accomplished by the older mechanists by means of face and spur geering, the acting teeth being disposed and modified, so that they should pass at the instant of contact in the same relative positions as if the axes had been parallel.

Another form of toothed wheel, known as "Hooke's Geering," from the name of the inventor, Dr. Hooke, is depicted in the annexed figures. This consists in disposing the teeth upon the peripheries of the wheels in equidistant steps, as shown in B, or in such manner as to form a continuous slope, as in C. The reason of the disposition will appear more fully when we come to describe the action of the teeth of one wheel upon those of another; but, in the meantime, the principle may be indicated by stating that the object aimed at is to reduce the rubbing friction as far as possible. In simple rolling contact, when the line of contact is in the plane of centres, the motion is entirely free of all rubbing friction; but toothed wheels, as will afterwards be shown, cannot fulfil this condition. The rubbing, however, may be diminished, by diminishing the size of teeth; but this is only practicable within the limits of the requisite strength of the teeth. To attain the same object without diminution of strength, Dr. Hooke had recourse to the contrivance described—which he denominates, in one of his Cutlerian lectures, "the perfection of wheel work," and states that he submitted a description of the invention to the Philosophical Society in 1660. This remarkable contrivance, which has several times since been re-invented and patented, was intended, to use the words of the learned inventor, "First, to make a piece of wheel-work so that both the wheel and pinion, though of never so small a size, shall have as great a number of teeth as shall be desired, and yet neither weaken the work nor make the teeth so small as not to be practicable by any ordinary workman. Next, that the motion shall be so equally communicated from the wheel to the pinion, that the work being well made, there

Figs. 19.



can be no inequality of force or motion communicated. Thirdly, that the point of touching and bearing shall be always in the line that joins the two centres together. Fourthly, that it shall have no manner of rubbing, nor be more difficult to be made than the common way of wheel-work, save only that workmen have not been accustomed to make it."*

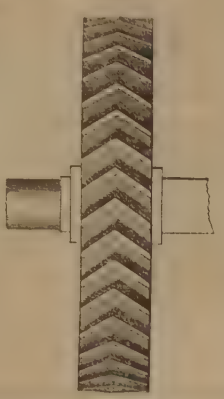
Without remarking further on these conditions, than that the fourth, of no rubbing, is necessarily included in the third, it may be observed that the mode of construction contemplated by Dr. Hooke was to make the wheel and pinion of several plates, laid one beside the other, and to bolt them together, as shown in the preceding figures. In the front elevation A is shown a wheel and pinion, formed each of six equal thicknesses of metal. These plates are individually cut into wheels which are fitted together in the manner described, but in such order that the teeth of the successive plates follow each other in regular gradation, so that the last tooth of each group may, within one step, answer to the first tooth of the next group. Thus reckoning from *a* to *b*, in the edge view B, the teeth follow each other in equidistant steps of such magnitude that *b* is distant one such step from *c*, the first tooth of the next group. The pinion being constructed in the same manner, and of the same number and thickness of plates, it is obvious that the inequalities in the touching surfaces are reduced six times, the number of thicknesses being six; and the amount of rubbing friction is reduced to what it would be between the two next teeth of one of the sets, that is, also six times. In effect, the inequalities and rubbing friction of a wheel of 36 teeth would thus be reduced to about the same as in a wheel of 36 × 6 = 216 teeth.

This is still further reduced by reducing the steps formed by the ends of the teeth, as shown in B, to a straight, or rather a spiral edge, as shown in C. A wheel and pinion of this kind, observes Dr. Hooke, are equivalent to the same having an infinite number of teeth, and are best made each of a single plate of convenient thickness, "which thickness must be more or less, according to the bigness of the sloped tooth. And this is always to be observed in the cutting of the teeth, that the end of one sloped tooth on the one side be full as forward as the beginning of the next tooth on the other side;" in other words, that the end *b* of one tooth on the right side be full as low as *c*, the beginning of the next tooth on the left side.

In a pair of wheels of this construction it is easy to perceive that the contact of the teeth will bear every instant at a single point, which, as the wheels revolve, will travel from one side to the other, a fresh contact always beginning on the first side immediately before the last contact has ceased on the opposite side. The contact, moreover, being always in the plane of the centres of the pair, the action is reduced to that of rolling; and as there is no sliding motion, there is consequently no rubbing friction between the teeth.

The motion of wheel-work of this description is remarkably smooth and free of vibration, but is liable to the objection of introducing endlong pressure upon the axes, in consequence of the obliquity of the surfaces of contact to the planes of rotation. To obviate this objection, the wheels have been made with double sloped teeth, in the form represented in the annexed figure. This wheel may be conceived to be composed of two thicknesses of metal, each having teeth formed upon its periphery in the same manner as the wheel C of figure 19, and that these plates are fitted together with the direction of the teeth reverse to each other. More strictly the wheel may be supposed to consist of two equal and concentric plates, which being, in the first place, made as separate wheels, have their teeth of equal size, and cut at equal but contrary angles with the axis of rotation, are fitted together so that the teeth of the one plate meet those of the other plate in the plane of contact. The slopes of the teeth on the two sides being thus reverse, and projected at equal but contrary angles with the axis of the wheel, and the pinion being formed in the same manner, the action of the respective sides of the V-shaped teeth on each other will be as before described; but the endlong pressure arising from the obliquity of contact on one side, will be exactly equal and opposite to that resulting from the obliquity of action on the contrary side; these pressures must, therefore, neutralize each other, and relieve the journals of an amount of friction which is necessarily involved in the mode of action contemplated in the original invention.

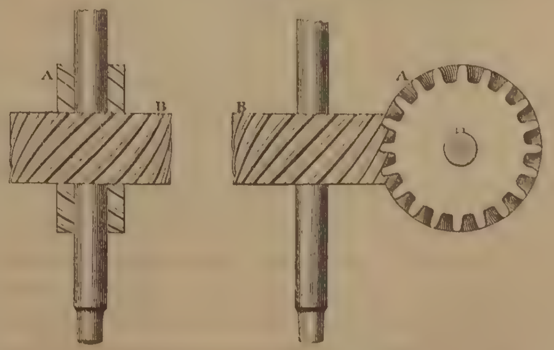
Fig. 20.



This modification was first proposed by Mr. White of Manchester, who obtained a patent for it in 1808. He endeavoured in vain to introduce it into practice, notwithstanding its manifest advantage where smoothness of action is required. The principal objection to which it is subject, consists in the difficulty of casting wheels of that pattern; this being surmounted, we know of no system of geering which possesses the same delicacy of action; and, in small machinery, it must be considerably more durable than any other variety of wheel-work, on account of the total absence of all abrading action.

A further modification of Dr. Hooke's geering has of late been somewhat extensively adopted, especially in the newer cotton-spinning machines. This consists, when the direction of the motion is simply to be changed to an angle of 90°, in forming the teeth upon the peripheries of the pair at an angle of 45° to the respective axes of the wheels; it will then be perceived that if the sloped teeth be presented to each

Figs. 21.



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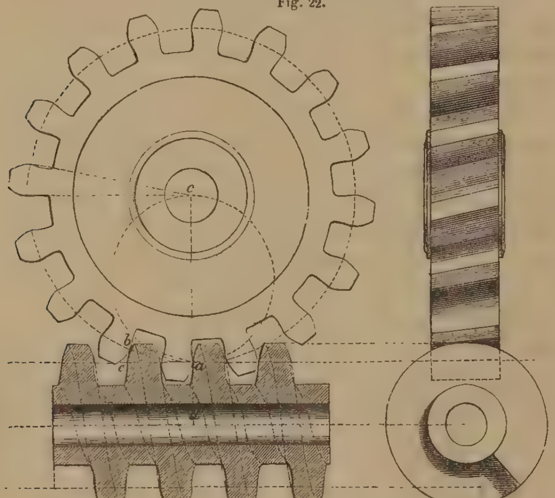
* Cutlerian Lectures, No. 2, by R. Hooke, Camb. 1674

other in such a way as to have exactly the same horizontal angle, the wheels will geer together, and motion being communicated to one axis, the same will be transmitted to the other at a right angle to it, as in a common bevil pair. Thus, if the wheel A upon a horizontal shaft have the teeth formed upon its circumference with an angle of 45° to the plane of the axis, it can geer with a similar wheel B upon a vertical axis. Let it be upon the driving shaft, then the motion transmitted will be changed in its direction, as if A and B were a pair of bevils of the ordinary kind, and as with bevils generally, the direction of motion will be changed through an angle equal to the sum of the angles which the teeth of the wheels of the pair form with their respective axes.

The objection in respect of endlong pressure manifestly applies to this arrangement with the same force that it does to the original inclined-plane wheel of Dr. Hooke; and if the action be downwards upon the step of the upright shaft, supposing the horizontal shaft to be the driver, this defect will be greatly aggravated. In practice, however, this is rarely the case in machinery of good construction; acting, therefore, in the opposite direction, the evil is reduced, at least as respects friction, and the arrangement is decidedly preferable to skew-bevils, which it would be necessary to adopt to accomplish the purpose for which wheels of this construction are commonly introduced.

The contrivance as stated is directly a modification of Dr. Hooke's geering; but, in effect, it may be described as a modification of the tangent screw and screw-wheel. In this last arrangement, represented by annexed fig. in its common form, it will be observed that the plane of the thread coinciding so nearly with that of the axis of the wheel, renders it necessary that the screw be uniformly the driver; and as the screw, by one revolution, passes only a single tooth of the wheel, the motion is necessarily slow. Both of these circumstances are manifest advantages in a large class of instances; but for very many purposes it is desirable to retain the principle of the screw, with an increase of velocity and a diminution of its rubbing friction. This is accomplished by diminishing the inclination of the thread of the screw to the axis—more correctly by increasing the number of separate spiral threads upon the surface of its cylinder; for, as every one of those spirals will pass its own wheel-tooth across the line of centres in a revolution of the screw, it follows that as many teeth of the wheel will pass that line during one revolution of the screw as this last has threads. If, therefore, we suppose the number of threads to be increased until they equal in number the teeth of the wheel, then the screw and wheel may be made exactly alike. This is precisely the case exemplified by fig. 21, in which we suppose A and B to be the same size, and to have the same number of spiral teeth—which might manifestly be continued round the axes of the wheels without affecting in the least their mode of action.

Fig. 22.

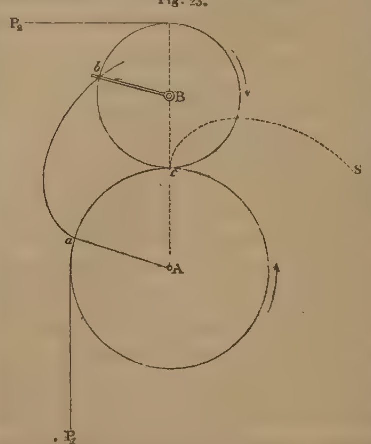


SECTION II.—RELATIVE DIRECTION AND VELOCITY OF ROTATION.

It is necessary in entering upon the consideration of the transfer of motion by geering, to remark that the path of movement of any revolving part is assigned by its connection with the frame-work on which it is supported; but its velocity, and the direction of its rotation, are defined by its connection with the other parts of the train to which it belongs. Thus, a wheel describes a circle because its axis is fixed; but the direction and the velocity of its rotation depend upon its connection with the next wheel of the train which lies between it and the moving power. As respects the direction of rotation, this is abundantly obvious without further illustration, and it is scarcely less evident that the relative velocities of any given pair, must depend upon the relative magnitudes of the circles which they respectively describe. Thus, confining our attention to the action of a wheel upon its pinion, supposing the circumference of the wheel be double that of its pinion, and, therefore, to have double the number of teeth upon it, we readily perceive that every revolution of the wheel must of necessity effect two revolutions of the pinion about *its* axis. And similarly knowing the respective circumferences of any pair acting together, it is easy to assign the ratio of their velocities. This form of the question, however, more rarely comes into use than that requiring the determination of those quantities from the diameters, which is slightly more involved, and renders it necessary, before entering further upon the consideration of the subject, to examine the conditions a little more closely.

Supposing that A and B in the annexed diagram, fig. 23, are two axes of motion referred to the same perpendicular plane, and that A B is a right line connecting them. Further, let it be supposed that the moving force is transmitted through A, and that it is required to connect it with B in such a way—that is, by a pair of spur-wheels of such sizes—that while A makes two revolutions B shall make three revolutions, the distance between A and B being six feet. The question is manifestly independent of the actual or absolute velocities of rotation of the pair, that is, of the number of times they shall revolve in any given unit of time.

Fig. 23.



General Solution.—To render the conditions of the problem as distinct as possible, let it be observed that whatever may be the relative diameters of the wheels, their lines of circumference must meet somewhere in the line AB, which is the shortest path between the axes. It is also manifest that the circumferences of the wheels, however different the circles they describe may be in radius—the axes being fixed, and the circumferences in *geer*—must of necessity have the same velocity. Thus, supposing for an instant that the question is already resolved, and that the circles described about A and B, having their point of contact at *c*, define the circumferences of wheels of the required sizes, then if a constant force, P_1, act tangentially to the circle described about A with the radius A*c*, it is evident that the rotation of this circle being transmitted to that described about B with the radius B*c*, will cause it to describe an equal *length* of arc to that through which itself passes, in any unit of time. In other words, the amount of rotation measured in lineal units upon the circumferences, is constantly the same as between *ca* and *cb*. But although the arcs be of the same absolute length, it is easy to perceive that they form very different portions of their respective circles. By an easy process of reasoning it is, indeed, shown that the arcs described by the two points of *a* and *b*, taken as portions of the circle to which they belong, are in the inverse ratio of the radii A*c* and B*c* by which the circles are described: that is,

the arc *ca* : the arc *cb* : : rad. B*c* : rad. A*c*.

Now, if this be true for a part of a revolution, it is equally true for a whole revolution of either of the wheels, and consequently for any number of revolutions. Hence we conclude that the *number of revolutions* of the point *a* about A are to those of the point *b* about B, as the *rad.* B*c* is to the *rad.* A*c*. But the relative number of the *revolutions* is obviously the same as the *relative velocity* of the pair, which is, therefore, expressed by the proportion above stated. And, generally, if two bodies revolve in circles and describe equal spaces in the same unit of time—that is, have the same velocity of rotation—then are their *angular velocities* measured as arcs of their respective circular orbits in the constant ratio of the radii of these orbits. That is, calling V the angular velocity described by the radius A*c* and measured by the arc *ca*, and *v* the angular velocity described by the radius B*c* and measured by the arc *cb*, then we have the following analogy:

$$\left.\begin{array}{l} V : v :: Bc : Ac \\ v : V :: Ac : Bc \end{array}\right\} \text{that is, } \frac{V}{v} = \frac{Bc}{Ac} \text{ or } \frac{v}{V} = \frac{Ac}{Bc}$$

And since the angular velocities are directly as the number of revolutions of the wheel and its pinion performed in the same time, it follows, from the preceding analogy, that the radii of the pair are in the inverse ratio of the number of revolutions which they severally make about their fixed axes.

To obtain, therefore, the radii of the wheels—which is the subject matter of inquiry—it only remains to divide the distance between the centres—that is, the line AB into parts inversely proportional, either to the angular velocities of the pair, or to the number of revolutions which the wheels respectively make in the same unit of time. That is, if for simplicity we call R the radius of the wheel, *r* the radius of the pinion, and let V and *v*, as before, express the number of revolutions which the wheel and pinion perform in the same time, then the analogy

$$\frac{V}{v} = \frac{Bc}{Ac} \text{ becomes } R \times V = r \times v$$

which, expressed in words, is, that the product formed by multiplying the radius of the wheel into its angular velocity (or number of revolutions in the unit of time), is equal to the product formed by multiplying the radius of the pinion into its angular velocity (or number of revolutions in the unit of time). Hence, therefore, the

$$\left.\begin{array}{l} \text{radius of the wheel } R = r \times \frac{v}{V} \\ \text{radius of the pinion } r = R \times \frac{V}{v} \end{array}\right\} \ldots\ldots\ldots\ldots (1)$$

Again, the distance of the centres being equal to the sum of the radii, if we call the distance *d*, we have manifestly

$$d = R + r \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

These expressions contain, besides the ratios of the velocities, *three* quantities; and, therefore, if two of these quantities be known the other two may be found by elimination between the equations. Thus, let the distance *d*, and the relative number of revolutions of the wheel and pinion be given, as in our question, and we have

$$R = \frac{d \times v}{V + v} \qquad r = \frac{d \times V}{V + v}$$

These rules expressed in words are as follows:—

I. To find the radius of the wheel, the relative number of revolutions and the distance between the centres of the pairs being given, multiply the distance between the centres by the numbers expressing the velocity of the pinion, and divide the product by the sum of the numbers expressing the relative velocities of the pair—understanding by velocities the number of revolutions which the wheels make in the same time.

II. To find the radius of the pinion or other wheel—multiply the distance between the centres by the number expressing the velocity of the wheel, and divide the result by the sum of the velocities; or subtract the radius of the wheel from the distance between the centres, the remainder is manifestly the radius of the pinion, that is, $r = d - R$.

Now in the arithmetical question proposed, the distance *d* between the centres is six feet, the velocity of the wheel two, and that of the pinion three, hence

$$R = \frac{d \times v}{V + v} = \frac{6 \text{ feet} \times 3}{2 + 3} = \frac{18 \text{ feet}}{5} = 3 \text{ feet } 7\tfrac{1}{5} \text{ inches.}$$

$$r = \frac{d \times V}{V + v} = \frac{6 \text{ feet} \times 2}{2 + 3} = \frac{12 \text{ feet}}{5} = 2 \text{ feet } 4\tfrac{4}{5} \text{ inches.}$$

$$\text{Sum of radii } d = R + r = 6 \text{ feet.}$$

To ascertain whether this solution be arithmetically correct, let us reverse the operation, and from the given radii R and *r* of the pair, determine the relative velocities of the axes A and B.

The radius of the wheel on A being 3 feet $7\frac{1}{5}$ inches, the diameter is 7 feet $2\frac{2}{5}$ inches = 7.2 feet; and the circumference of a circle being 3.1416 times its diameter, the circumference of the wheel is

$$7.2 \text{ feet} \times 3.1416 = 22.62 \text{ feet.}$$

Similarly the radius of the pinion on B being 2 feet $4\frac{4}{5}$ inches, the diameter is 4 feet $9\frac{3}{5}$ inches = 4.8 feet, and the circumference

$$4.8 \text{ feet} \times 3.1416 = 15.08 \text{ feet.}$$

Now, as these circumferences have the same velocity, it is evident that the number of revolutions of their respective axes must be inversely as the numbers 22.62 and 15.08. That is, supposing the wheel upon A to make one revolution, then must a point in the circumference of the pinion on B describe likewise a circular path equal in length to the circumference of the wheel on A. But this last is 22.62 feet, whereas the circumference of the pinion is only 15.08 feet. In order, therefore, that the velocity may be the same, a point taken in the circumference of the pinion must make one whole revolution and 7.54 feet more; that is (15.08 + 7.54 feet) = 22.62 feet. But 7.54 feet is half of 15.08 feet, consequently, in travelling over 22.62 feet, the circumference of the pinion necessarily makes one revolution and a half; and the same is manifestly true of the axis upon which it is placed. The ratio of the circumferences of the wheel and pinion being therefore 1.5 to 1, the ratio of the velocities of the axes is as 1 to 1.5, which may be expressed by the following analogy:

cir. of pinion : cir. of wheel : : revol. of axis of wheel : revol. of axis of pinion,

which applies to any pair. Taking the numbers given in the example discussed, we have

$$15.08 : 22.62 :: 1 : \frac{22.62}{15.08} = 1\tfrac{1}{2}.$$

And the ratio of 1 to $1\frac{1}{2}$ is the same as 2 to 3, the condition announced in the question proposed.

Having the radii or diameters of the wheels given, it is not, however, necessary to find their circumferences in order to determine the ratio of the velocities of their axes. We have here adopted that mode for the sake of illustrating more clearly the truth of the rules deduced by a general consideration of the question proposed; but assuming a very slight knowledge of the mathematical properties of the circle, it may be shown that

$$\left.\begin{matrix}\text{Rad.}\\ \text{Diam.}\end{matrix}\right\}\text{of pinion} : \left.\begin{matrix}\text{Rad.}\\ \text{Diam.}\end{matrix}\right\}\text{of wheel} :: \text{Revol. of axis of wheel} : \text{Revol. of axis of pinion.}$$

Thus, in our question the radius of the pinion is 2 feet $4\frac{4}{5}$ inches, and that of the wheel 3 feet $7\frac{1}{5}$ inches.

$$\text{Therefore}\left\{\begin{array}{l}\text{2ft. }4\tfrac{4}{5}\text{ ins.} : \text{3 ft.}7\tfrac{1}{5}\text{ ins.} :: 1 : \dfrac{\text{3 ft. }7\tfrac{1}{5}\text{ ins.}}{\text{2 ft. }4\tfrac{4}{5}\text{ ins.}} = 1\tfrac{1}{2}\\ \text{4ft.}9\tfrac{3}{5}\text{ ins.} : \text{7 ft.}2\tfrac{2}{5}\text{ ins.} :: 1 : \dfrac{\text{7 ft. }2\tfrac{2}{5}\text{ ins.}}{\text{4 ft. }9\tfrac{3}{5}\text{ ins.}} = 1\tfrac{1}{2}\end{array}\right.$$

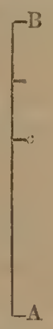
as before found. Consequently, if the diameters or radii of a pair be known, the relative number of revolutions which they will make in a given unit of time is ascertained by dividing the measure of the one by that of the other, the quotient being the *number of revolutions, or parts of a revolution, the axis of that wheel will make, which is made the divisor, while the axis of the wheel whose diameter or radius is divided makes one revolution.*

To express these rules in general language, let the angular velocities of the two axes be V and v, the radii of their two geering wheels R and r, diameters D and d, and circumferences C and c; then

$$\frac{V}{v}=\frac{r}{R}=\frac{d}{D}=\frac{c}{C} \text{ and } \frac{v}{V}=\frac{R}{r}=\frac{D}{d}=\frac{C}{c}$$

and these ratios are constant, whatever be the absolute velocity of the driving wheel.

Questions of this kind may be solved very readily by means of the compasses and a scale of equal parts. Thus let A and B be the given centres, and let the ratio of their velocities be respectively *two* and *three*, as before; then, if the line joining the centres A and B be divided into 2 + 3 = 5 equal parts, that is into as many equal parts as there are units in the terms of the given ratio, the radius of the wheel upon A will contain three of these parts, and the radius of the pinion on B will contain the remaining two parts, and the point of contact of the wheels will be at c. Thus, the line joining A and B being 6 feet, each of the five divisions is 1 foot $2\frac{2}{5}$ inches. Therefore



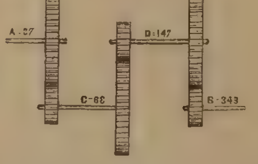
$$\left.\begin{array}{l}Ac = 1\text{ ft. }2\tfrac{2}{5}\text{ ins.} \times 3 = 3\text{ ft. }7\tfrac{1}{5}\text{ ins.}\\ Bc = 1\text{ ft. }2\tfrac{2}{5}\text{ ins.} \times 2 = 2\text{ ft. }4\tfrac{4}{5}\text{ ins.}\end{array}\right\}\begin{array}{l}\text{the radii as be-}\\ \text{fore found.}\end{array}$$

This method is very convenient in practice when the terms of the ratios are small; but when the numbers are high, time is commonly saved by adhering to the arithmetical rule already stated, by which the point c will always be obtained by a single application of the compasses.

In the preceding examples, only a single pair has been taken; but it frequently occurs in practice that motion is to be transferred through a system of shafts, with varying degrees of intensity, according to the purposes to be attained. It likewise not unfrequently happens that a pair of shafts are to be connected at such distances as to require the interposition of *carrier* wheels, that is, wheels intended simply to connect the two, without being subservient to any other purpose. This, however, when the intermediate distance is considerable, is usually and most conveniently attained by bevil geer; but as we shall presently see, the principle is the same, and therefore, in the meantime, it may not be inappropriate to give a general example illustrative of the ease and readiness with which such questions may be resolved, without having recourse to the mode of trial and error so commonly resorted to in practice.

Let there be two motion-axes, A and B, of which the angular velocities are as 27 to 343, and let it be required to connect them by four wheels on two intermediate shafts, the wheels to have a common velocity ratio throughout. Let A be the driver; then the intensity of the motion is required to increase from 27 towards B, in which it is expressed by 343.

Fig. 24.



Now, it has been shown that if a pair of wheels be in geer, their radii are to each other as the angular velocities, that is, the relative number of revolutions which they make upon their axes. In order, therefore, that the geering pairs in the question proposed may have a common velocity ratio, it is obvious that the same must apply to the axes upon which they are fixed. These velocities ought, therefore, to form a geometrical progression, of which the first and last terms, (A = 27 and B = 343), and also the number of terms namely, the four axes, are given. Now, the formula for all such questions, m being the number of geometrical means to be inserted, ϱ the ratio, is for an increasing series*

$$\varrho = \sqrt[m+1]{\frac{B}{A}}\text{; which becomes } \varrho = \sqrt[3]{\frac{343}{27}} = \frac{7}{3}.$$

$$\text{Hence the angular velocity of } C = 27 \times \frac{7}{3} = 63.$$

$$\ldots\ldots\ldots\ldots\ldots\ldots \text{ of } D = 63 \times \frac{7}{3} = 147.$$

Also, A will be connected with C, and C with D, and D with B by wheels of radii 7 and 3 respectively.

It might, however, have so happened that it was required to bring up the speed, say at D, to 216; and, in that case, the uniform ratio of the connecting wheels would be thus interrupted. The question would, indeed, be reduced to this —required to find the angular velocity of C, that of A being 27, and of D = 216.

There being only one intermediate term, it is a mean proportional, and can be found at once by extracting the square root of the product of the two given terms, that is, by finding $\sqrt{216 \times 27} = 76.367$, which expresses the angular velocity of C, and, of course, also the radius of the wheel upon A, that upon C geering with it being 27.

Had we applied our formula, we should have found

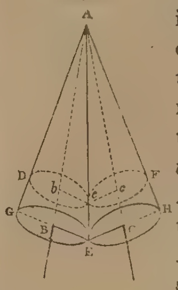
$$\varrho = \sqrt[m+1]{\frac{D}{A}} = \sqrt{\frac{216}{27}} = 2.8284.$$

And, therefore, C = 27 × 2.8284 = 76.367 as before. The former calculation is, however, done with greater facility.

These examples illustrate the whole process of calculation of the transfer of angular velocity from one axis to another, and simple and common place as they do appear, they include every case of the kind to be met with in practice.

In the case taken, we have supposed the motions of the pair to be in the same plane, and the axes of rotation of the circles parallel. But we have already seen that these conditions are not constant, and that it is found necessary to change the direction of motion through all conceivable angles to meet the exigencies which arise in practice. We have also seen that two cases of change of direction present themselves: in the one case the axes of rotation meet in a point as in common bevils and face wheels, and in the other the axes are neither parallel nor do they meet in direction—a circumstance to which is chiefly owing the use of skew-bevils and screw-wheels. These cases require to be separately noticed.

Fig 25.



Let us, in the first place, suppose that the two axes of rotation BA and CA intersect in A, and include the angle BAC. If we suppose a cone to be generated by the revolution of the line AE about AB, and another by the revolution of the line AE about AC, then these cones being made to revolve *in contact* about their respective axes AB and AC, their surfaces will *roll* upon one another along their whole length of contact AE, and no part of the surface of one would slide upon that of the other, and therefore the whole surface of one cone, which passes in a given time over the line of contact AE, must be equal to the whole surface of the other, which passes that line in the same time. For, as already shown, if n times the circumference of a circle De be equal to m times that of a circle Fe, and these circles be conceived to revolve in contact about their centres b and c, and to carry the cones with them, then it is evident that whilst the cone EAG makes n revolutions the cone EAH will make m revolutions. But n times the circumference of any other circle, EG of the one cone is equal to m times the circumference of the corresponding circle EH of the other cone; for the diameters of these circles, and, therefore, their circumferences, are to one another in the same proportion as the diameters and circumferences of the circles eD and eF. It is therefore obvious, that whilst the cones make n and m revolutions respectively, the circles EG and EH are carried through n and m revolutions respectively, and that n times the circumference of EG being equal to m times the circumference of EH, it follows that these circles *roll* in contact through the whole of their path. And the same is equally true of any other corresponding circles in the cones, and therefore of their whole surfaces, so that the rotation of one axis being communicated to the other, by the rolling together of the two cones, the surface of the one cone will carry with it the surface of the other along the whole line of contact AE with equal perimetral velocities, and with angular velocities inversely proportional to the circumferences, diameters, or radii, of the corresponding circles. If, therefore, V and v be used to represent the angular velocities, or number of revolutions, of the axes AB and AC, then will

$$\frac{V}{v} = \frac{ce}{be} = \frac{CE}{BE} = \frac{r}{R}$$

which is the same constant ratio as that found when the axes were parallel; and as in that case, it is independent of the absolute velocity of the driving cone.

In practice, *thin* frusta only of the cones, with teeth upon *their* perimeters, are employed; but in this there is no new consideration involved, as respects the angular velocities of the axes upon which the wheels are carried. In determining the size of a pair of bevils, we are not, however, limited to any particular diameters as when the axes are parallel; the wheels may be made of any convenient sizes, and the teeth consequently of any breadth, according to the stress they are intended to bear—a circumstance of the utmost practical importance, because, by this method, they admit of being made of any required degree of strength, and as we shall afterwards have occasion to explain, their action is essentially accompanied with less friction, and is more smooth and uniform than that of spur geer.

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* A being the first term and B the last, the series expressed generally is of the form

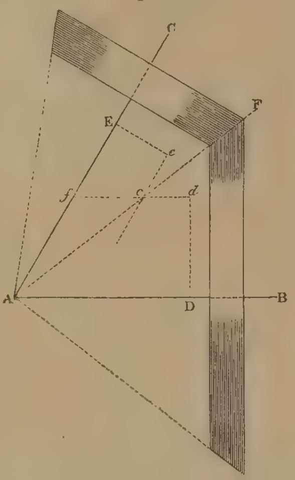
$$A + A\sqrt[m+1]{\frac{B}{A}} + A\sqrt[m+1]{\frac{B^2}{A^2}} + A\sqrt[m+1]{\frac{B^3}{A^3}} + - - + B$$

And which when $\frac{B}{A}$ is not a whole number, may be expressed more conveniently,

$$A + \sqrt[m+1]{A^m B} + \sqrt[m+1]{A^{m-1} B^2} + \sqrt[m+1]{A^{m-2} B^3} + - - + B$$

The question, however, which presents itself here, is the mode of determining the *relative* sizes of the conical frusta of a pair; and this resolves itself into a division of the angle included between the two axes inversely as the ratio of their angular velocities. The mode of accomplishing this is simple. Let B and C be the position of the two given axes, and let them be prolonged till they meet in a point A. Further, let it be required that C make *seven* revolutions, while B makes *four*. From any points D and E in the lines AB and AC, and perpendicular to them, draw D*d* and E*e* of lengths (from a scale of equal parts) inversely as the number of revolutions which the axes are severally required to make in the same unit of time. Thus, the angular velocity of axis B being 4, and that of the axis C being 7, the line D*d* must be drawn = 7, and the line E*e* = 4. Then through *d* and *e* parallel with the axes AB and AC draw *dc* and *ec* till they meet in *c*. A straight line drawn from A through *c* will then make the required division of the angle BAC, and define the line of contact of the two cones, which fulfil the assumed conditions of angular velocity and inclination of their axes, and in which two frusta may therefore be applied at any suitable distance from the common apex A. The ratio of the pair is thus the ratio of the lines D*d* and E*e* directly.

Fig. 26.



The point *c* may also be found more directly thus: From A towards C in the axis AC, set off from a scale as many equal parts (A*f*), as there are units in the number (7) expressing the velocity of that axis; from the point *f* draw *fc* parallel to AB, and set off from the same scale as many parts (*fc*), as there are units in the number (4) expressing the velocity of the axis AB; then a line drawn from A through *c*, as before, will divide the angle, as required.*

By one or other of these methods the division of the angle of inclination of the axes may always be determined, the ratio of the angular velocities of the pair being known. From the same data the angles at the vertices may also be found, though not in a form to be of much practical use in the workshop.†

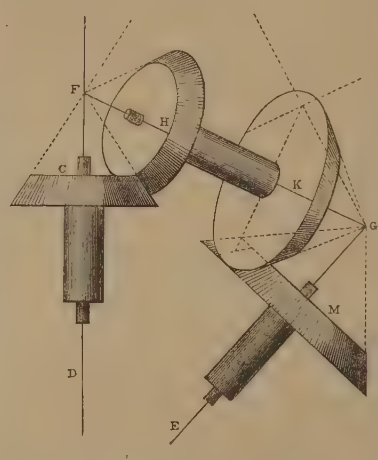
Having determined the *relative* sizes of the wheels, the next object is to determine their *absolute* sizes, and this will depend upon the power they are required to transmit, and other circumstances, as velocity and pitch, to which we shall return.

We have, in the preceding example, supposed the axes to include less than a right angle, in order that our solution might be sufficiently general; when the angle of the axis is exactly 90 degrees the solution is still the same, but, if possible, more simple and palpably obvious. The principle embodied is, that the perimeters of every corresponding pair of circles move in the same direction at the point of contact, although the circles are not in the same plane; and this is evident since the radii of contact are perpendicular to the line of intersection of the two planes which contain the circles, and consequently the tangents at the point of contact coincide with that line, and with each other.

The case in which the axes are neither parallel nor intersecting, admits of solution by means of a pair of bevils upon an intermediate axis, so situated as to meet the others in any convenient points.

Thus, if DF and EG be the two given axes, they may be connected by a third axis FG, intersecting them in the points F and G; and if a shaft be mounted in the position of this third axis with a pair of bevils upon it, geering with the bevils on the main axes, and having their apices in the points F and G of these axes, the motion of the driving shaft will be communicated, as if the two bevil wheels C and M were in immediate contact;—the ratio of their velocities remaining the same, provided the bevils H and K be of the same size. If it be required to bring up the speed or reduce it, between the two shafts in a higher ratio than is convenient by that arrangement, the interposed bevils will afford additional facility of accomplishing the purpose.

Fig. 27.



In the preceding figure the pairs are of such relative sizes as only slightly to modify the speed; but taking another case under the same circumstances and supposing EG to be the driving shaft, and to have an angular velocity of 3; if, then, it be required that the shaft DF have a corresponding velocity of 34, and that the speed be brought up uniformly, the radii of the wheels, taken pair and pair, will have a common ratio, determined in accordance with the angular velocities of the shafts upon which they are carried.

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* For $\frac{fc}{Af} = \frac{\sin fAc}{\sin Acf} = \frac{\sin fAc}{\sin cAB} = \frac{CF}{BF}$.
But *fc* expresses the angular velocity of the axis B, and A*f* that of the axis C, quantities which we formerly denoted by V and *v*, therefore,

$$\frac{fc}{Af} = \frac{V}{v} = \frac{CF}{BF} = \frac{r}{R},$$

which is the velocity ratio required.

† Thus if θ be the angle of the axes BAC, and τ the semi-angle at the vertex of the cone on AC, that is, the angle CAF, and V and *v* the terms of the angular velocities, as above stated, then

$$\frac{\sin \tau}{\sin (\theta - \tau)} = \frac{V}{v}\ ; \text{ whence } \tan \tau = \frac{\sin \theta}{\frac{v}{V} + \cos \theta}.$$

To adapt this formula to logarithms take a subsidiary angle ϕ such that $\cos \phi = \frac{V}{v} \cos \theta$;

$$\text{whence } \tan \tau = \frac{V \sin \theta}{2\, v \cos^2 \frac{\phi}{2}}.$$

If θ be a right angle, as is very commonly the case in practice, then

$$\tan \tau = \frac{\sin \theta}{\frac{v}{V} + \cos \theta} = \frac{V}{v}.$$

That this condition may be fulfilled, it is necessary that the speed of the intermediate shaft be a mean proportional to the speeds of the other two: that is,

$$\sqrt{34 \times 3} = 10.0995,$$

the angular velocity of the intermediate shaft. Now the radii of the wheels being inversely as the angular velocities of the shafts which they connect, will be thus expressed,

$$\text{cone M} : \text{cone K} : : 10.0995 : 3;$$
$$\text{cone C} : \text{cone H} : : 34 : 10.0992,$$

that is, a common ratio of 3.33317 to 1, and this, as already explained, applies to the radii, diameters, and circumferences of the wheels.

When the contiguity of the shafts is such as to permit of their being connected by a single pair, skewed bevils are frequently employed, as shown in fig. 18, with teeth cut in the direction of the line of contact of the two surfaces. In strictness these wheels ought to be frusta of hyperbolic conoids, that is, of solids formed by the revolution of hyperbolæ about their axes; but as in practice the frusta include a very small portion of the surface curves, they are usually made plain. The error thereby introduced is not of sensible importance, and to avoid it would not compensate for the additional labour of attaining the higher degree of accuracy. This will fall to be fully discussed when we come to treat of the teeth of wheels; and as respects the relative velocities of a pair of this kind, it is evident that the same law obtains as in the preceding cases. For whatever may be the obliquity of direction of the teeth, the angle cannot in any way affect the length of perimeters, that is, the amount of surface which passes over the line of contact in a given unit of time.

When the axes are at right angles to each other, and do not intersect, the wheel and screw may be employed to connect them. The velocity of angular motion is in this arrangement immediately deduced from that of the screw, its number of threads, and the number of teeth in its geering wheel. Thus, if it be required to transmit the motion of one shaft to another contiguous, and at right angles to it—the angular motions being as 20 to 1; then, if the screw be a single-threaded one, the wheel must have 20 teeth; but if double-threaded the number of teeth will be increased to 40, for 2 teeth will be passed at every revolution. If the velocities be as 2 to 1, the condition is, that the screw have half as many threads upon its barrel, as there are teeth on the wheel; and if 1 to 1, the wheel and screw lose their distinctive characters: both become many-threaded screws under the form of wheels, as represented by fig. 21,—which may be taken as an illustration of the disguise that a common arrangement may assume when carried to an extreme. Wheels of this sort may often be applied with peculiar advantage, especially in light geering; and when so applied it is not essentially necessary that the axes be at right angles to each other any more than it is in bevil geer.

If the screw have few threads compared with the number of teeth of the wheel, it must always assume the position of driver on account of the obliquity of the thread to the axis; and in this respect its action is analogous to that of a travelling rack, moving endways one tooth, whilst the screw makes one revolution on its axis.

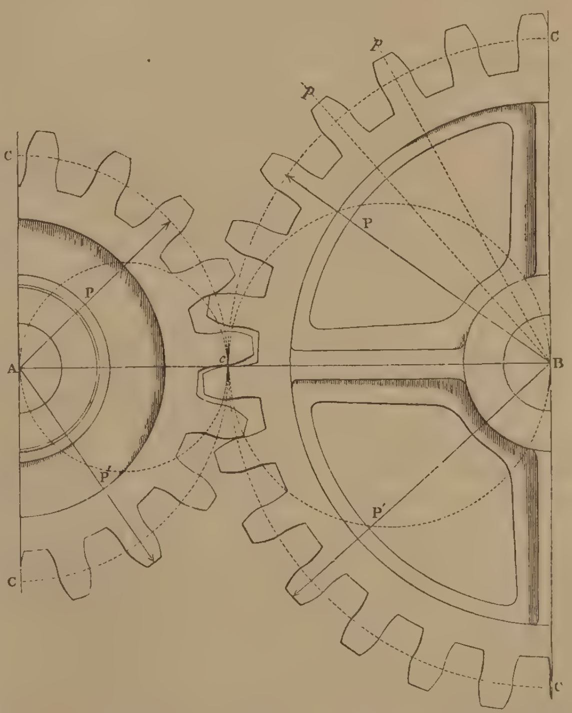
In relation to this subject generally, it may be here observed, that the velocity ratio of the geering ought uniformly to be adjusted so that the motive power shall work at its best speed—that is, the speed at which it will yield the maximum of useful effect. Any other arrangement must manifestly be attended with loss of power. It is also of importance that the first mover be connected as directly as possible with the working points, and that the requisite speed be brought up as near as possible to it when ranges of shafting are necessarily interposed. By that means the geering is lightened, the friction and vibration diminished, and to that extent the power is economised.

The preceding calculations respecting the relative angular velocities of combination of wheels may be rendered still more obvious and familiar, by reference to the numbers of teeth in the geering wheels. But in order that this may be fully understood, it is necessary to define more precisely the conditions involved in the working together of a pair of toothed wheels.

SECTION III. — ON THE PITCH OF WHEELS.

The primary object aimed at in the construction of toothed geer is the uniform transmission of the power, supposing that to be constant and equal. This implies that the one wheel ought to conduct the other as if they simply touched in the plane passing through both their centres. This plane is denoted by the line AB, in the annexed figure—which is here introduced for the purpose of illustrating the few terms which we shall have occasion to employ in describing the actions of toothed geer. When this line—which is usually denominated the *line of centres*,—is divided into two parts, Ac and Bc, proportional to the number of teeth formed upon the perimeters of the pinion and wheel, these two parts are *proportional* or *primitive radii* of the pair; and a circle CC being described from each centre passing through

Fig. 23.



the common point *c*, limits what is called the *pitch line* or *circle;* that is, a circle described from the centre A, and another from the centre B, through the same point, are called, the first, the *pitch circle* or *pitch line* of the pinion, and the other of the wheel. They are also sometimes called the *primitive* and *proportional* circles. If the pitch circle be divided into as many equal parts as there are teeth to be given to the wheel, the length of one of these parts is termed the *pitch of the teeth.* One of these arcs, as that intercepted by *p p* in the figure, comprehends a complete *tooth and space*, meaning by *space* the hollow opening between two contiguous teeth.*

By the pitch lines of a geering pair is, therefore, to be understood the proportional circles in which they would revolve upon each other if they were simply cylinders without teeth; and the pitch of the teeth is the length of arc of the pitch circles, measured from the centre of one tooth to the centre of the contiguous one. Any right lines, P and P, drawn from the centre of the wheels till they meet the pitch circles or lines, are termed proportional radii, because they determine the relations of their angular velocities; and any similar radial lines, P′ and P′, continued to the extremities of the teeth, are called the *true radii* of the wheel and pinion —for no very obvious reason. In bevil and conical wheels the pitch circle is the circumference of the base of the frustum, as AB of Fig. 1. Plate V.

It is scarcely necessary to observe that the teeth being equidistantly disposed upon the circumference of a wheel, it is necessary that the pitch include an arc of such length that $N \times P = C$, in which N is used to denote the number of teeth, P, the pitch, and C, the circumference of the wheel. Knowing the circumference, it is also obvious that $P = \frac{C}{N}$. Thus the circumference C being = 100 inches, and the number of teeth $N = 40$, then the pitch $P = \frac{100}{40} = 2\frac{1}{2}$ inches.

Although C cannot always so readily be attained, yet, knowing from the properties of the circle that $C = \pi D$, in which D expresses the diameter multiplied by a constant quantity, $\pi = 3.1416$, that is, nearly $\frac{22}{7}$, we are provided with an equivalent expression nearly as simple, namely, $P = \frac{\pi D}{N}$; and from this, if any two of the quantities P, N, D, be given, the third may be found.

Thus, giving $P = \pi \times \frac{D}{N}$, whence $D = N \times \frac{P}{\pi}$ and $N = \frac{\pi}{P} \times D$.

These rules expressed in words are as follows:—

I. To find the pitch of the teeth of a wheel, the diameter and number of teeth being given, divide the diameter D, (in inches,) by the number of teeth N, and multiply the quotient by 3.1416: the product is the pitch in inches or parts of an inch. Or multiply the diameter by 3.1416, and divide the product by the number of teeth. As an example, let the diameter be $31\frac{4}{5}$ inches, and the number of teeth 50; then $31\frac{4}{5} \div 50 = 0.636$; and $0.636 \times 3.1416 = 2$ inches, the pitch.

II. To find the diameter of a wheel, the number of teeth and pitch being given, divide the pitch by 3.1416, and multiply the quotient by the number of teeth. Or first multiply the number of teeth by the pitch, then divide the product by 3.1416. Let the number of teeth be 60, and the pitch $2\frac{1}{2}$ inches; then $2\frac{1}{2}$ inches $\div\ 3.1416 = 0.7958$ and $0.7958 \times 60 = 47.75$ inches nearly.

III. To find the number of teeth, the diameter and pitch being given, divide 3.1416 by the pitch, and multiply the result by the diameter in inches. Or first multiply the diameter by 3.1416, then divide by the pitch. Let the diameter be 30.5 inches, and the pitch half an inch, then $3.1416 \div 0.5 = 6.2832$ and $6.2832 \times 30.5 = 191.6376$. But the question does not admit of fractions, consequently we must either increase or diminish the diameter so that the number of teeth shall either be exactly 191 or 192. Assuming that the last is more convenient, then we have, by the second rule,

$$D = (192 \times .5) \div 3.1416 = 30.557 \text{ inches diameter.}$$

These rules, together with those for the calculation of the relative velocities and diameters of wheels, comprehend every case that can occur in practice; but calculations of this kind are commonly referred as far as possible to the pattern-book, in which the pitches are usually limited to inches and definite submultiples of an inch, especially in spur geer. In ordinary geering the pitches most commonly in use range from 1 inch to 4 inches, increasing up to 2 inches by *eighths*, and beyond by *fourths* of an inch. Below 1 inch, the pitches decrease by *eighths* down to $\frac{1}{4}$ inch; but wheels of so fine pitch are rarely required, and few pattern-books have much variety of them. In machine shops, on the contrary, wheels below 1 inch pitch are in most frequent requisition, and the pattern-book contains the greatest variety of them.

The pitches being few and definite, the rules given above may be greatly simplified by calculating beforehand the values of $\frac{P}{\pi}$ and $\frac{\pi}{P}$ corresponding to them. The following table is

---

* Some writers define the pitch to be the *chord* of the pitch arc, that is, the straight line subtending the arc *pp*. In wheels having 20 teeth, and upwards, the two definitions may be supposed to coincide, the difference between the arc and its chord being little more than $\frac{1}{1000}$ part of the radius. Thus if N be the number of teeth, and the radius of the wheel be taken as unity, we have

$$\text{pitch chord} = 2 \sin \frac{180^\circ}{N}\,;\ \text{pitch arc} = \frac{2\,\pi}{N}.$$

If $N = 20$ then $\frac{180^\circ}{N} = 9^\circ$, the sine of which is .15643; therefore

$$2 \sin \frac{180^\circ}{N} = 2 \times .15643 = .31286\,;\ \frac{2\,\pi}{N} = \frac{6.2832}{20} = .31416.$$

Then $.31416 - .31286 = .0013$; consequently, if the wheel have a radius of 9 inches, the difference is .0117 inch, and .234 inch upon the circumference of 20 teeth.

For a wheel of 40 teeth the difference is .00016 of radius, which is little more than an *eighth* of the difference with 20 teeth. In practice, therefore, the error arising from measuring the pitch, as a chord of the pitch arc, may be diminished by supposing the wheel to have double or triple the number of teeth; for by this means the pitch arc will be divided into a corresponding number of parts, and the sum of the chords will not differ sensibly from the length of the arc. This precaution is requisite in laying off pinions, for as the leaves become fewer than 20 the difference becomes speedily appreciable.

In defining the pitch to be the *arc*, the notion of rolling contact—which is the aim of all our investigations respecting the best form for the teeth of wheels—is better preserved, and, moreover, the phraseology is simplified. It is true, that the motion of the smoothest working wheels is still a stepping motion; but in practice it answers the end of purely rolling contact, and therefore may be reasoned upon and described, as if the purpose were fulfilled in theory, as it is in practice.

constructed to answer the purpose, and will be found very convenient when the diameter D is to be determined, the pitch P and number of teeth N being given, and conversely, when the diameter and pitch are given to find the number of teeth.

	$D = \frac{P}{\pi} \times N$	$N = \frac{\pi}{P} \times D$
Pitch in inches and parts of an inch.	RULE.—To find the diameter in inches, multiply the number of teeth by the tabular number answering to the given pitch.	RULE.—To find the number of teeth, multiply the given diameter in inches by the tabular number answering to the given pitch.
Values of P	Values of $\frac{P}{\pi}$	Values of $\frac{\pi}{P}$
6	1.9095	.5236
5	1.5915	.6283
4½	1.4270	.6981
4	1.2732	.7854
3½	1.1141	.8976
3	.9547	1.0472
2¾	.8754	1.1333
2½	.7958	1.2566
2¼	.7135	1.3963
2	.6366	1.5708
1⅞	.5937	1.6755
1¾	.5570	1.7952
1⅝	.5141	1.9264
1½	.4774	2.0944
1⅜	.4377	2.2848
1¼	.3979	2.5132
1⅛	.3568	2.7926
1	.3183	3.1416
⅞	.2785	3.5904
¾	.2387	4.1888
⅝	.1989	5.0266
½	.1592	6.2832
⅜	.1194	8.3776
¼	.0796	12.5664

The use of this table is very evident, and the manner of applying it may be rendered still more obvious by the following examples:—

1. Given a wheel of 88 teeth, 2½ inch pitch, to find the diameter of the pitch circle. Here the tabular number in the second column answering to the given pitch is .7958, which multiplied by 88 gives 70.03 for the diameter required.

2. Given a wheel of 5 feet (60 inches) diameter, 2¾ inch-pitch, to find the number of teeth. Here the factor in the third column corresponding to the given pitch is 1.1333, which multiplied by 60 gives 68 for the number of teeth.

It may, however, so happen that the answer found in this manner contains a fraction—which being inadmissable by the nature of the question, it becomes necessary to alter slightly the diameter of the pitch circle. This is readily accomplished by taking the nearest whole number to the answer found, and finding the modified diameter by means of the second column. The following case will fully explain what is meant:

3. Given a wheel 33 inches diameter, 1¾ inch-pitch, to find the number of teeth. The corresponding factor is 1.7952, which multiplied by 33 gives 59.242 for the number of teeth, that is, 59¼ teeth nearly. Now, 59 would here be the nearest whole number; but as a wheel of 60 teeth may be preferred for convenience of calculation of speeds, we may adopt that number and find the diameter corresponding. The factor in the second column answering to 1¾ pitch is .557, and this multiplied by 60 gives 33.4 inches as the diameter which the wheel ought to have.

Palpably simple as these operations may appear to a mathematician, they require more arithmetic than is always at the command of the workman, and accordingly mechanical modes of avoiding them are in constant use in the pattern-shop. The best of these we shall describe when treating of the construction of wheel patterns, and, in the meantime, proceed to explain a mode of sizing wheels in relation to their pitches, diameters, and numbers of teeth, which is adopted in some engineering factories as a simplification of that explained above.

Suppose the diameter of the pitch circle to be divided into as many equal parts as there are teeth to be given to the wheel; let one of these parts be called the *diametral pitch*, to distinguish it from the *circular pitch*, which we have hitherto employed; and let a few definite values (in terms of the inch) be assigned to it; then it is clear that calling M the diametral pitch, we have the relation $\frac{D}{N} = M$. And as M is always a simple fraction of an inch, let $M = \frac{1}{m}$, then we have the general expressions,

$$N = m \times D; \qquad \frac{N}{D} = m; \qquad D = \frac{N}{m}.$$

To illustrate this by an arithmetical example, let it be assumed that a wheel of 20 inches diameter is required to have 40 teeth; then the diametral pitch,

$$M = \frac{D}{N} = \frac{20}{40} = \frac{1}{m} = \tfrac{1}{2} \text{ inch};$$

that is, the diameter being divided into equal parts corresponding in number to the number of teeth in the circumference of the wheel, the length of each of these parts is ½ inch, consequently $m = 2$; and according to the phraseology of the workshop, the wheel is said to be one of *two pitch.* The circular pitch corresponding to this diametral pitch is by the properties of the circle $\frac{1}{2} \times 3.1416 = 1.5708$ inch.

In this mode of sizing wheels a few determined values are given to m, as 20, 16, 14, 12, 10, 9, 8, 7, 6, 5, 4, 3, 2, 1, which includes a variety of pitches from ⅕ inch up to 3 inches, according to the following table, which shows the value of the circular pitches corresponding to the assigned values of m.

Values of m.	1	2	3	4	5	6	7	8	9	10	12	14	16	20
Corresponding circular pitch in decimals of an inch.	3.142	1.571	1.047	.785	.628	.524	.449	.393	.349	.314	.262	.224	.196	.157

From this table, having the value of m given, the corresponding circular pitch is found; and from the rules given above, if the number of teeth and value of m be known the diameter of the wheel is also known, for $D = N \div m$. Thus if the number of teeth be 80 and $m = 10$, then the diameter $D = 8$ inches and the circular pitch is .314 inch, that is, $\frac{5}{16}$ inch very nearly. Generally, the diameter and value of m being given, the number of teeth is found from the rule $N = m \times D$. Thus the value of m being 10, and the diameter 20 inches, the number of teeth is 200.

From these remarks, it is easy to perceive that this mode of sizing wheels differs from that before explained simply in this, that it expresses in small whole numbers the quantity $\frac{\pi}{P}$ instead of the quantity P. In other words, the circular pitch being regarded as the circumference of a circle, the diametral pitch will be the diameter of that circle. Thus, in the example given above, the diametral pitch is $\frac{1}{2}$ inch, and the corresponding circular pitch is 1.5708 inch, which is the circumference of a circle whose diameter is $\frac{1}{2}$ inch. This mode of defining the pitch has hitherto been confined to the sizing of small wheels.

We observed in concluding Section II. that the calculation of the angular velocities of combinations of wheels is facilitated by reference to the numbers of teeth which they severally possess. The operation is simple, and may be dismissed with a short exposition of the general principle in connection with the preceding subject of the section.

In the first place, it is manifest that the numbers of the teeth must be whole numbers: the question cannot embrace fractions, for the fractional tooth would be really a tooth of smaller size than the others, and would have no property, save that of rendering the divisions of the wheel unequal and of causing shocks in the geering.

Although the reason has not as yet been explained, it may be observed in connection with the subject under consideration, that when a wheel and pinion geer together, their limiting ratio differs according as the wheel is the driver or follower, and that a smaller pinion may be employed to drive than to follow. Thus supposing the teeth to be perfectly formed, and that the *action commences at the line of centres*, the least wheel that can drive a pinion of 12 teeth is 30; but the same pinion would drive a wheel of 19. Again, nothing less than a rack would drive a pinion of 10 teeth, but that pinion would drive a wheel as low as 23 and upwards. No pinion of less than 10 can consequently be driven, but a pinion as low as 6 will drive a wheel of 176 and upwards. Lastly, the least pair of equal pinions that will work together is 16.

In practice, the teeth are formed so that the contact begins in advance of the line of centres, and therefore it is possible slightly to exceed these limits; still it is not desirable unless there be some very important object to be attained by limiting the ratio. This will be better understood when we have discussed the action of a geering pair in relation to the form of their teeth.

It has long been regarded as a rule among millwrights that the number of teeth in a wheel should be prime to the number of teeth in its pinion; in other words, that the number of teeth of the wheel should not be divisible by the number of teeth in the pinion without a remainder; and that the best possible relation of the numbers is such, that in effecting the division the remainder be 1. This *one* is termed a *hunting tooth*, and effects the purpose of preventing the same pair of teeth of the wheel and pinion from coming together until the former has made as many revolutions as it has teeth. Thus in a pair of 49 and 16 teeth, the same teeth will come together only once during 49 revolutions of the wheel; whereas, had the numbers been 48 and 16, the same teeth would act together every third revolution of the wheel. By such an arrangement it is supposed the wear would be less uniform; and it may be observed, that if the teeth be at first incorrectly made, there is some advantage to be gained by taking prime numbers; for the tendency of the teeth acting together is to grind themselves into the proper shape, and by distributing their original irregularities over the whole, instead of bringing them more frequently together, the grinding process is effectually diffused and ultimately an approximation to the true form of the teeth is attained.

This precaution was undoubtedly of advantage before attention was given in practice to the proper form of the teeth of wheels; when the teeth were in fact nothing better than rectangular blocks, hardly regular in form, and often irregularly disposed upon the periphery of the wheel. But in the practice of the present day, when millwrights are fully alive to the method and advantages of giving to the teeth, in the construction of their wheel patterns, the proper geometrical form, and do not trust to the wheels wearing themselves into shape, the precaution of making the numbers of the wheel and pinion prime to each other, is less required and may, in fact, be disregarded in proportion as accuracy of construction is attained.

In respect of the relative sizes of the pairs which geer together, it is further to be observed, that these may be employed for two purposes. In the first place, it may be desired simply to transfer the motion of one axis to another without change of velocity, and without changing the direction of the motion. This is commonly effected by the interposition of a wheel upon a third axis, otherwise unconnected with the system. But the main purpose to be accomplished is the modification of the contemporary velocities of the parts to such extent that their respective speeds shall be adapted to the work to be performed at the several points. The former class of wheels do not form any part of the data upon which a calculation of the numbers of the geering depend, and may therefore be overlooked, and the pairs they connect be reckoned as if in actual contact, but having the direction of the communicated motion preserved by the carrier wheel which is interposed.

To exhibit the method of applying the principles of angular velocity, explained in a preceding section, to the computation of the numbers of a system of toothed geering, we shall consider, in the first place, the action of a single pair. The fundamental proposition may be stated thus:—If there be an equal pair in geer, then whether the pinion drive the wheel or be driven by it, the number of turns of the wheel multiplied by the number of its teeth, is equal to the number of turns which the pinion makes in the same time, multiplied by the number of its teeth, so that the number of the contemporary turns of the wheel and pinion are reciprocally proportional to the numbers of their teeth. In other words, if the number of the teeth of the wheel W and the pinion P be represented by these letters, and the numbers of their contemporary revolutions by the small letters w and p, then is $w \times W = p \times P$, and consequently $w : p :: P : W$. For the number of the teeth of the wheel being represented by W, at each turn made by the wheel a certain number of the teeth represented by W will be engaged by the pinion. While the wheel therefore makes a certain number of turns represented by w, it will be engaged in the pinion with a certain number of teeth represented by $w \times W$. And since P represents the number of teeth of the pinion, at each turn made by the pinion it will be engaged in the wheel with a certain number of teeth represented by P. While the pinion, therefore, performs the certain number of revolutions expressed by p, it will be engaged in the wheel with a number of teeth represented by $p \times P$. But while the wheel and pinion perform their contemporary revolutions, as many teeth of the wheel will be engaged in the pinion as there will be teeth of the pinion engaged with the wheel; it is therefore clear that $w \times W = p \times P$, and considering the two members of this equation as the products of the extremes and means of a proportion, we have $w : p :: P : W$, which is the proposition announced.

And since $w \times W = p \times P$ we manifestly have $w = \frac{p \times P}{W}$ and $p = \frac{w \times W}{P}$.

That is, 1st, The number of revolutions of the wheel is equal to the product of the number of contemporary revolutions, and of the number of teeth of the pinion, divided by the number of teeth of the wheel. And 2d, The number of revolutions of the pinion is equal to the product of the number of contemporary revolutions, and of the number of teeth of the wheel, divided by the number of teeth of the pinion.

Again, when the wheel makes only one revolution, that is, when $w = 1$, the number of revolutions of the pinion will be equal to the number of the teeth of the wheel, divided by the number of teeth of the pinion; for on that supposition $p = \frac{w \times W}{P}$ becomes $p = \frac{W}{P}$.

And when the pinion makes one revolution, that is, when $p = 1$, the number of revolutions of the wheel will be equal to the number of leaves of the pinion, divided by the number of the teeth of the wheel; for we shall then find that the expression $w = \frac{p \times P}{W}$ becomes $w = \frac{P}{W}$.

What is thus true of one pair, is equally true of any number of pairs. For example, when in a system of geering a wheel W geers with a pinion P, and a second wheel W′, upon the same shaft which carries that pinion, gives motion to a second pinion P′; a third wheel W″ on the shaft of this second pinion acts upon a third pinion P″; a fourth wheel W‴ on the shaft of the third pinion moves a fourth pinion P‴; and so on to any number of pairs; if the numbers of the teeth of the wheels and pinions be denoted by the large letters, and the number of the contemporary revolutions of the wheel W, and the pinions P, P′, P″, P‴, by the corresponding small letters, w, p, p', p'', p''', we shall have the following proportions:

1st. The contemporary revolutions of the first pair, namely, wheel W, and pinion P, being represented by w and p, we have, $w : p :: P : W$.

2d. The contemporary revolutions of the second pair, namely, pinion P′, and wheel W′, being represented by p, and p', we have, $p : p' :: P' : W'$.

3d. The contemporary revolutions of the third pair, namely, pinion P″, and wheel W″, being represented by p', and p'', we have, $p' : p'' :: P'' : W''$.

4th. The contemporary turns of the fourth pair, namely, pinion P‴, and wheel W‴, being expressed by p'', and p''', we have, $p'' : p''' :: P''' : W'''$.

And multiplying all these proportions in order, we find: $w : p''' :: P \times P' \times P'' \times P''' : W \times W' \times W'' \times W'''$.

From which we readily deduce the expression:

$$p''' = w \times \frac{W \times W' \times W'' \times W'''}{P \times P' \times P'' \times P'''}$$

In words, the number of revolutions of the last pinion of the system, is equal to the number of revolutions of the first wheel multiplied by the quotient which is found by dividing the continued product of the numbers of teeth of all the wheels, by the continued product of the numbers of teeth of all the pinions.

It thence also follows that for one turn of the first wheel W, that is, when $w = 1$, the number of revolutions of the last pinion will be denoted by

$$p''' = \frac{W \times W' \times W'' \times W'''}{P \times P' \times P'' \times P'''}$$

In words, the number of revolutions of the last pinion, for one revolution of the first wheel, is equal to the product of all the wheels divided by the product of all the pinions.

When the wheels have all the same number of teeth, and the pinions likewise have all the same number; if n be the number of wheels or pinions in the train, then we shall manifestly have for the number of revolutions of the last pinion during a revolution of the first wheel, the expression $p''' = \left(\frac{W}{P}\right)^n$.

These are the cases which commonly occur in practice; but it may also be required to determine the inverse condition; that is, having the number of revolutions of the last pinion given, it may be required to determine the corresponding speed of the first wheel. This is at once found, from the proportion above stated, to be

$$w = p''' \times \frac{P \times P' \times P'' \times P'''}{W \times W' \times W'' \times W'''}.$$

And if $p''' = 1$, this becomes, as in the preceding case,

$$w = \frac{P \times P' \times P'' \times P'''}{W \times W' \times W'' \times W'''}.$$

Which may be expressed verbally, by saying, that for one revolution of the last pinion, the first wheel will make that portion of a revolution expressed by a fraction, having for its numerator the product formed by multiplying together the numbers of the teeth of all the pinions, and for its denominator the product formed by multiplying together the numbers of teeth of all the wheels. It is in fact the fraction found by inverting the terms of the expression for p'''.

From these rules it immediately follows, that whether a system of geering contains one wheel and pinion, or any greater number of wheels and a like number of pinions, if we designate the product of all the wheels by W, and the product of all the pinions by P, and if p represent the number of revolutions made by the last pinion during one revolution of the first wheel, we shall have $W = p \times P$ and $p = \frac{W}{P}$.

As the number of the teeth of each wheel ought to be without a fraction, it follows that the product W of all the wheels must be a whole number; and the product $p \times P$, which is equal to W, will also be a whole number. It is therefore obvious, that if p, which represents the number of revolutions made by the last pinion, during a revolution of the first wheel, contains a fraction which cannot become a whole number, unless it be multiplied by a number equal to its denominator, the product of the pinions P, which is always a whole number, must be equal to the denominator of that fraction, or a multiple of it. Thus if in the expression $p \times \mathrm{P}$ we find $p = 7\frac{50}{51}$, then P must be 51, or some multiple of that number, as the double, triple, and so on, of 51.

We have here spoken of a system of geering, as if uniformly composed of a series of wheels and pinions in regular succession; this is rarely the case, as the power is commonly required to be led off at various points, with velocities suited to the particular operations to be performed. The principle, however, remains unaffected by any deviation from the regular order assumed; and may be further generalized by regarding the distinction of wheels and pinions employed above, to imply merely that of two series of wheels, those of the first order convey the motion which they receive to those of the second order. Instead, therefore, of understanding the calculations to be restricted to the particular case, although this be the most common, in which the speed is brought up from the prime mover, onwards as far as the power is conveyed, we may conceive that the term *wheels* employed above, and designated by W, to include all the terms, whether wheels or pinions, or both, of the odd numbers of the train, 1, 2, 3, 4, 5, 6, and so on, namely, 1, 3, 5, and that the term *pinions* similarly employed, includes all the even terms of the same train, namely, 2, 4, 6; there being six wheels in the system, of which those denoted by the odd numbers, 1, 3, 5, are drivers, and those denoted by the even numbers, 2, 4, 6, are driven.

To illustrate these observations by a simple example, let us take that of the Thrashing machine, shewn in Plates IX. and X. In this example, the water wheel, which is the prime mover, is 16 feet diameter, and makes seven turns per minute; upon its shaft is fixed the spur wheel marked I, of 121 teeth, which geers with a pinion marked 3, of 16 teeth on a second shaft. The speed, therefore, of this second axis is $\frac{7 \times 121}{16} = 52.94$ turns per minute. Upon this second shaft is also fixed the spur wheel, marked 2, of 115 teeth; and this last geers with the pinion 4 of 15 teeth, on the end of the beater shaft. The speed of this last is therefore $\frac{52.94 \times 115}{15} = 405.94$ turns per minute of the beaters.

This might be found at once from the expression,

$$p'' = w \times \frac{\mathrm{W} \times \mathrm{W}'}{\mathrm{P} \times \mathrm{P}'} \text{ which becomes } 7 \times \frac{121 \times 115}{16 \times 15} = 405.94;$$

and consequently, for each revolution of the water wheel shaft, or of the first wheel 1 of the train, this last pinion 4 makes

$$\frac{\mathrm{W} \times \mathrm{W}'}{\mathrm{P} \times \mathrm{P}} = \frac{121 \times 115}{16 \times 15} = 57.993 \text{ revolutions.}$$

The ratio of their velocities is therefore 1 to 58, very nearly.

Again, the shakers S are driven by a pitch chain from the second shaft, which, we have seen, has a speed of 52.94 turns a minute, and the driving pulley is half the diameter, and consequently has upon its circumference half the number of studs (which answer the purpose of teeth in this case) that the driven pulley has; the speed of this last is therefore $\frac{52.94}{2} = 26.47$ turns per minute. The shaft of the second set of shakers being driven off this last by equal pulleys, its velocity is consequently the same. The feed-rollers are also driven by a pitch chain from the second shaft; the pulley on this last is to that of the feed-rollers, (not shown in the drawings) as 6 to 5; the velocity of the rollers is therefore $\frac{6 \times 30.25}{5} = 63.528$ turns per minute.

The fanners, again, are driven from the beater-shaft by a belt which passes over the guide pulleys *b*, *b*, to the fanner shaft *c*; the pulley upon this last is to that on the beater-shaft, as 7 to 6, and consequently the velocity is $\frac{405.94 \times 6}{7} = 347.949$ turns per minute.

These last might of course be calculated directly, by application of the general expression for the velocity of the last wheel of a train, the velocity of the first wheel and number of teeth of the intermediate wheels being known.

When bands are employed to transfer motion from one axis to another, the diameters of the pulleys determine the relation of the angular velocities; and the numbers expressing those diameters are accordingly employed in the calculations relating to speed, in exactly the same manner as the numbers of the teeth are employed, when the axes are connected by toothed geer. Bands are commonly resorted to only when the speeds are high: they are seldom employed to transmit any considerable power at a velocity of less than a foot a second. It is also often required to effect a sudden and great reduction of velocity at some point in the general system for some particular purpose of a secondary kind to the main object of the geering. When this occurs, recourse is frequently had to the tangent screw and wheel, as in general the simplest of the class of contrivances applicable to purposes of this nature; and in that case, the data of calculation are the number of threads on the screw—which may be single, double, or triple threaded—and the number of teeth on the wheel upon which the screw acts. As before explained, a single threaded screw is to be considered as a wheel of one tooth; when double threaded, as a wheel of two teeth, and so on; but it is not often that any greater number of threads are cut upon screws intended to be employed in this manner.

SECTION IV.—ON THE STRENGTH OF THE TEETH OF WHEELS.

In reference to the strength of the teeth of wheels, the first subject of inquiry is the *stress* they are severally required to sustain when in action.

The fundamental principle involved in the consideration of this subject, is expressed by the condition, that *the pitch being the same, the stress is inversely as the velocity;* and this is obviously true, since the teeth which act with superior force must be proportionally stronger; and the momentum of the power remaining constant, the higher the velocity becomes, the more is the weight of the power diminished, so that in any combination of wheels, the stress upon the teeth is reciprocally proportional to the velocity at a given point of the train. Thus for example, if the pitch circles of one pair of wheels move at the rate of 10 feet in a second, and another pair of wheels, under precisely the same circumstances, transmit the same amount of power at a velocity of 5 feet in a second, the stress on the latter is double that on the former pair.

Conversely, the strength which is sufficient to transmit a given amount of horse power, when the velocity is 1 foot per second, will be equal to the transmission of double that amount when the velocity is 2 feet a second; three times the amount when the velocity is tripled; and generally *n* times the horse power, when the velocity becomes *n* times greater. Knowing therefore the strength of teeth necessary to transmit a given amount of horse power, we know that the same strength of wheel will be sufficient under any other circumstances of increase, or diminution of velocity, when the horse power of the first mover in both cases, divided by the velocity in feet per second, produces the same quotient. Thus assuming that a wheel having teeth $1\frac{1}{4}$ inch thick, by 9 inches breadth, is sufficient in strength to sustain 24 horse power, when the velocity of its pitch circle is 3 feet per second, then according to the principle explained, the same wheel will do for 48 horse power, when its velocity is 6 feet a second; and will only be equal to the transmission of 8 horse power, when the velocity is decreased to 1 foot a second, the quotient of the horse power by the velocity being in these instances constantly 8.

This proposition, taken absolutely, is true only of the wheels of the same train; but it is rendered universal, by determining a standard of velocity and effect, that is, of mechanical power, by which comparison may be made, of that developed under any variable circumstances which may arise. Such standard is furnished in the received mechanical unit of a horse power, namely, 33,000lbs. raised 1 foot high in a minute; otherwise expressed, this is equivalent to 550lbs. raised 1 foot in a second. Hence, then if H be the number of horses' power of any first mover, and v the velocity of the pitch circle (in feet per second) of any wheel in the system of geering moved by it, then the stress will be expressed in lbs. by $\frac{550 \times H}{v}$.

For example, if the pitch line of a wheel move with a velocity of 11 feet per second, and the power of the prime mover be 20 horse power, the stress will be $\frac{550 \times 20}{11} = 1000$lbs.

Otherwise, if P be the power of the prime mover in pounds, and V be the velocity of that power in feet per second, the stress on the teeth of a wheel through which the power is transmitted, and of which the velocity of the pitch circle is v, will be expressed in lbs. as before, by $\frac{P \times V}{v}$.

It is necessary, however, to be observed, that the absolute power of the prime mover must only be considered at those points of the geering through which it is wholly transmitted; for if the power be taken off at different points, it is obvious that the stress will be successively diminished as these points are passed. For instance, the power of a steam-engine being employed to drive a cotton factory; if the first geering be so arranged that the whole power of the engine is transmitted from the fly-wheel shaft to a vertical shaft, which ascends from the bottom to the top-flat of the building, by a bevil pair, and if the geering of the several flats be successively connected with this upright shaft, it is clear that, in estimating the stress at the several points, with a view to ascertaining the requisite strength of the several pairs, the whole power of the engine ought to be taken only at the first point, that is, at the point where it is connected with the vertical shaft. In estimating the strength of the bevil pair there placed, H in our formula will be equivalent to the whole power given off by the engine; but at the successive points, where the power is taken off to drive the machinery of the several flats, H will represent only the amount of power requisite to do the work at these points. Let us suppose that the factory contains four flats, and that the engine works to 60 horse power; if the work done in each flat be the same, then the value of H, at each of the four points where the geering of the flats is connected with the driving shaft, will be 15 horse power, for although 60 horse power really passes into the main shaft, a fourth of that power is given off at the first point of connection, another fourth at the second point, a third fourth at the third point, and the remaining fourth is taken up at the fourth point. Suppose then the shaft to be composed of four lengths coupled together, the first length ought to be of sufficient strength to transmit 60 horse power, the second 45 horse power, the third 30 horse power, and the fourth 15 horse power. But this does not apply to the wheels by which the power is thus taken off; for each pair having an equal resistance, namely, 15 horse power, to overcome, their strength must be equal, supposing their velocities equal.

It is rarely, possibly never the case, that the amount of power is the same for each flat; the different operations which are conducted in a factory are usually so arranged, that more or less power is required to drive the machinery of a flat according as it is devoted to *preparation*, spinning, weaving, and also according to the kind of machines employed and the fineness of the yarns.

A difference is besides very properly made in practice, in the strength of those wheels of a system of geering which are placed near the first mover, to compensate for irregularities in the motion; for were the strength exactly limited to the resistance to be overcome under constant action, a sudden acceleration of the speed would tend to strip the wheels, or in other words, to break the teeth. Also in operations of an irregular kind, the strength ought to be greatly more than is requisite in such geering as that of a cotton factory. Thus the geering in iron-works, and the like, is greatly beyond the strength which a calculation of the power of the prime mover would indicate, and this is required to counteract the sudden shocks which result from the chocking of *rolls*, &c.

It may also be necessary to remark, that, in estimating the strain upon a system of geering, the nominal power of the first mover may be very different from the real power given out in doing the work. The nominal power of a steam-engine, for example, may not exceed a half of the power at which it is actually made to work; and we have known it to be less than a third. In calculations of this nature, it is therefore the actual power required to do the work which is to be taken as an element of the data—that is, the horse power at which the *resistance* is valued.

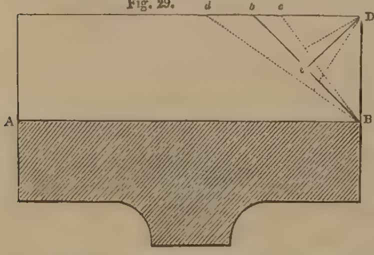
With these remarks we proceed to examine more closely the circumstances under which the strength of the teeth of wheels is to be determined in relation to the power they are required to transmit.

The ordinary mode of considering the subject is to regard the teeth as beams: the breadth of the wheels, the length from the root to the point, and the thickness of the teeth being supposed to represent the breadth, length, and depth of the beam.

This rule would undoubtedly be perfect, could we rely on wheels being made mathematically correct in the form of their teeth and in their fixings, and also were the shafts upon which they are placed absolutely free of all vibration; for then in working every point of the breadth of the tooth would bear its proper share of the whole strain. This degree of accuracy cannot however be wholly realized in practice, and every deviation from absolute correctness is attended with inequality of action for which the rule requires a corresponding degree of restriction in its application. Even were the first construction perfect, the unequal wear of the brasses and of the teeth themselves, arising from inequality in the hardness of the material and other causes developed under the particular circumstances of the work, the springing of the shafts and the like, would be sufficient to divert the wheels from the truth, and thereby cause the teeth to bear harder on some point than on another.

Now in the teeth of wheels, as in every other part of a mechanical structure, it is of importance that the whole be made of such strength, as not only to sustain uninjured the greatest stress that is likely to come upon it in the course of working in the best possible position, but also in the worst—that is to say, in the direction in which the structure is capable of offering the least effective resistance to fracture. Now supposing the strain to act with its whole energy upon the extreme corner of a tooth, it is easy to perceive that it will there more readily produce fracture than if it acted along the whole line of the breadth; for supposing it to act along the whole line of breadth, if fracture of the tooth should take place, it must traverse the whole line AB, or some line parallel to it; but acting at D, should the force be sufficiently great to break the tooth, the fracture will take place, not along the root of the tooth AB, but in the line B*b*, or in some line parallel to it—these being the lines of less resistance, under the condition of the pressure being applied at D, than any lines parallel to the base of the tooth.

Fig. 29.



To show that this is strictly true, let it be borne in mind that the strength of beams of equal thickness is directly as their breadths and inversely as their lengths; consequently, if the proportion of the length to the breadth be preserved, the strength will remain unaltered whatever difference is made in the actual dimensions. But this being true, it necessarily follows, that the line of fracture, under the circumstances presumed, will be along the line, which, as a base, bears the least proportion to the perpendicular height from that base to the point where the pressure is applied. This least proportion is when $BD = Db$, for the base Bb is then only double the effective length of the beam, that is, double the perpendicular $e\,D$. If a line be drawn from B to *d*, it will manifestly be more than double the altitude of the triangle $B\,d\,D$, of which it is the base; and, similarly, the line $B\,c$ is more than double the altitude of its triangle $B\,c\,D$.

This might be directly proved by application of the rules of *maxima* and *minima;* rather it might be shown that the strain is greatest under a force acting at D, in a line determined by $D\,B = D\,b$, which makes $B\,b = 2\,e\,D$.

In effect, therefore, the line $B\,b$ (including all lines parallel to it) is the line of least resistance of the tooth, and consequently the line in which fracture would be produced by a force sufficiently great applied at D. Presuming then that wheels, in consequence of inaccuracy of workmanship, unequal wearing of brasses, vibration of shafts and other circumstances incidental to the action of a system of geering, are liable to stress acting upon them in the least efficient position of the teeth, it would appear that the *effective* proportion of the breadth of the tooth, assuming the thickness to be uniform, does not exceed twice the length. The effect of the section of the tooth not being uniform, to alter the line of least resistance need not be taken into account, as the principle is not affected by the amount of alteration thereby induced. Whatever may be the force, the principle informs us that there is a limit beyond which no strength is practically gained, and this limit will be found in general not to differ materially from double the length of the tooth.

It must not, however, be inferred from this, that it is useless to make the breadth of any proportion greater than that stated; for although no additional practical strength be gained by increase of breadth, it is still highly advisable that the dimensions named should bear a much higher ratio, than is given by the consideration of the merely mathematical principle. This is accordingly followed in practice, and the advantage is, that the wearing action, by being distributed over a larger surface, does not so soon reduce the thickness, and thereby render the wheel too weak for the work it has to perform. Moreover, there is the additional advantage in giving more breadth than is indicated above, that the surfaces of contact being longer, vibration is to some extent diminished; the centres are accordingly better preserved, and the wearing of the tooth becomes greatly more uniform. An error of workmanship, and of unequal contraction of the casting, becomes likewise more apparent; and may possibly admit of correction; and should fracture actually take place in consequence of such fault, the reason will not be assignable to miscalculation of strength.

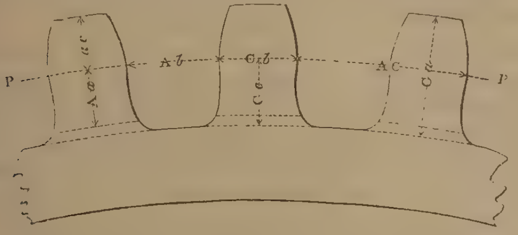
As experience may, on this head, have more weight than mere *a priori* reasoning, we shall subjoin a table in the proper place, of examples of wheels in actual use, showing the actual pitches and proportions of breadths and lengths adopted in the best modern geering. But in the meantime, to show more connectedly the fundamental conditions embodied in the relation of toothed geer to the work it is intended to perform, we shall assume that experience, in all its details, coincides with the maxims developed by experimental investigation of the strength of materials. The laws determined in that department of physical inquiry, form the code of ultimate appeal; and, conjoined with experience in the particular applications to which our attention is at present more immediately directed, afford sufficient guidance to the practical millwright and engineer, in proportioning the strength of his geering to the amount of stress which it is intended to sustain.

To bring the subject within precise limits, let us for the present assume that the form of the teeth is that of a rectangular prism, and this will in general be sufficiently accurate for practical purposes, if the least thickness be taken, and it enables us to apply with more freedom the rules of calculation. It has already been stated that the *true diameter* of the wheel is that which is measured from the extremities of the teeth in contradistinction to the diameter of the pitch circle; it therefore follows, that calling R the radius of the pitch circle, T the true radius, and D the projection of the tooth beyond the pitch circle, then $D = T - R$, a quantity which is by some conveniently termed the *addendum* of the teeth, and which in the practice of millwrights is commonly $\frac{3}{10}$ of the pitch. The part of the tooth from the pitch circle to the root, is in the same manner reckoned $\frac{7}{10}$ of the pitch; but of this depth $\frac{1}{10}$ is allowed for *clearance* at the bottom, so that only $\frac{9}{10}$ of the length of the teeth are brought into action.

As it is convenient to express all the dimensions in terms of the same unit, and the pitch being an appropriate quantity, is nearly universally adopted as the term of comparison; the thickness, and in consequence the width of space between, are similarly expressed. The thickness, it is evident, would equal *half* the pitch, if the spaces between the teeth were equal in width to the thickness of the teeth. But in order that the teeth of a geering pair may engage together and extricate themselves with facility, it is necessary that the space exceed the thickness of tooth by a small fraction, which though not often a constant part of the pitch, may be very nearly expressed by $\frac{1}{12}$,* thus giving as the thickness of tooth $\frac{5}{11}$ of the pitch, which leaves a width of space equal to $\frac{6}{11}$ of the same unit.

These proportions differ, though slightly, in different works and in different localities; but they are the most commonly employed, and are besides the most consistent with good and accurate workmanship. For the sake of more easy reference, we collect them into a table, which the annexed diagram will serve fully to explain. They stand thus:

Fig. 30.



$A\,C$ = Pitch of teeth = 1 pitch.

$a\,c$ = Depth to pitch line, PP, = $\frac{3}{10}$ ——.

$A\,a + a\,c$ = Working depth of tooth, = $\frac{6}{10}$ ——.

$C\,c - A\,a$ = Bottom clearance, = $\frac{1}{10}$ ——.

$C\,a$ = Whole depth to root, = $\frac{7}{10}$ ——.

$C\,b$ = Thickness of tooth, = $\frac{5}{11}$ ——.

$A\,b$ = Width of space, = $\frac{6}{11}$ ——.

These proportions, as remarked, are found to work very advantageously, and are those adopted in several workshops; but the following are preferred, or at least made use of by some engineers of experience. Thus supposing the pitch divided into 15 equal parts: then the

Depth from point to pitch line, = $5\frac{1}{2}$ parts.
Depth from pitch line to root of tooth, = $6\frac{1}{2}$ ——.
Whole length of tooth, = 12 ——
Working depth, = 11 ——.
Thickness of tooth, (also of arms and rim), = 7 ——.
Width of space, = 8 ——.

In practice, these proportions are usually laid off in lines for the convenience of the workmen in the pattern-shop, so that for any given pitch the other dimensions may at once be determined by means of the compasses, and without having recourse to calculation. In the annexed plate two diagrams of that sort are given. Scale 1 contains the proportions last enumerated, in which the pitch is supposed to be divided into 15 equal parts; and Scale 2 is constructed nearly according to the proportions first given, but embraces the recognised principle that the relative amount of clearance ought to vary inversely as the pitch, wheels of small pitch requiring more clearance relatively than those in which the pitch is greater. Accordingly, in this scale, the clearance in a wheel of $\frac{3}{4}$ inch pitch, is $\frac{1}{10}$ the pitch, whereas were the scale sufficiently extended, it would show a clearance of only $\frac{1}{18}$ for a pitch of 6 inches.

The construction of these scales is very simple. Thus in Scale 1, which is given full size, and to 6 inches pitch, let AB be divided into 15 equal parts, and draw BC perpendicular to it; and again divide BC into a determinate number of parts from B, actual measures of the pitches for which the scale is intended to be used; that is, $Ba = \frac{1}{2}$ inch; $Bb = 1$ inch; $Bc = 2$ inches, and so on, and join a and A; b and A; c and A; and so on. To complete the scale, draw 15 parallels to BC from the points numbered in the line AB, numbering their intersections (if thought proper), with the line AC in the same order; and also the 2 parallels T and U, (which are full lines in the diagram) equidistant from the parallels on each side of them.

The scale is thus ready for use, and its principle is self-evident. To get from it the several proportions for a given pitch, say of 3 inches $= Bd$, let the compasses be extended from the intersection of the parallel marked T, with the line AB, to the point where it intersects the line Ad; this will be the part of the tooth from the pitch line to the point, and equivalent to $5\frac{1}{2}$ parts of the pitch (viz. of Bd); similarly the compasses being extended from the intersection of the parallel U with the line AB, to its point of intersection of the line Ad, will give the part of the length of the tooth from the pitch line to the root, and equivalent to $6\frac{1}{2}$ parts of the pitch. For the whole length of the tooth (if wanted in one measurement) set the compasses to the point where the parallel marked 12 meets the line AB, and extend to its point of intersection of the line Ad at s; the length is 12 parts of the pitch Bd. The working depth is in like manner found from the parallel marked 11, the thickness from that marked 7 and the width of space from that marked 8.

The proportions for any other given pitch comprised in the scale are found in precisely the same way, and if the scale be well constructed they may be measured off with the utmost accuracy and readiness. To save confusion, it is, however, better in practice to insert in the diagram only those parallels, namely, T, U, 12, 11, 8, 7, which are required; the others are not requisite, and by inattention may lead to error. Both this scale and that marked 2, are commonly drawn on hardwood boards; but sometimes, for the sake of greater accuracy, on plates of polished brass.

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* Writers on the teeth of wheels commonly assign a *tenth* to a *fifteenth*, according to the "quality of the work," as the clearance; and a *tenth* in small wheels, and a fifteenth in those of large size, are very common proportions; but in wheels of 1 inch to 3 inches pitch, a twelfth is found to be enough and not too much.

The description of the scale as here given supposes that the lateral clearance is constantly $\frac{1}{15}$ of the pitch; but as it is commonly desired and desirable that this should vary slightly with the pitch, relatively increasing as the pitch decreases, two other lines *m n* and *p q* have been introduced into the scale, to enable such modification to be adopted should it be required. These lines are drawn at such angles as to give a clearance at 6 inches pitch of $\frac{1}{18}$, which is increased at $\frac{3}{4}$ inch pitch to $\frac{1}{10}$. From these lines the thickness and space are to be taken, instead of using the lines marked 7 and 8, setting the compasses in the points of intersection with the pitch lines, and extending perpendicularly to *the line* AB; in other words, the *shortest distance* from the point of intersection with the pitch line to the line AB, is the required measure of the space when the line *p q* is taken, and of the thickness of tooth when the line *m n* is taken.

Scale 2 is more complete than the one described, and when well constructed, insures, with moderate care, a degree of accuracy and uniformity in the construction of the various sizes of wheels for which it is employed, that can hardly be otherwise attained; and besides the accuracy of workmanship resulting from its use, it has the additional advantage of economizing the time of the workman. The principle of its construction is in effect the same as that described; but its use is more extended, the diameter of the wheel being found from it simultaneously with the length and thickness of tooth, width of space, and clearances. As here laid down, the scale is a *sixth* of the size of the original, and adapted to wheels of all the pitches enumerated in the table, page 83, from $\frac{1}{2}$ inch up to 3 inches. The mode of construction is this: having drawn the line AD of any convenient length, raise the perpendicular CB to it, also of any convenient length. On the line AD lay off the greatest pitch of the scale from C to A—(in this case $\frac{1}{6}$ of *three* inches). Then from C towards D lay off *seven* times the pitch *once* or *twice*, according to the sizes of wheels to which the scale is intended to be applied. In the scale given, double of seven times the pitch is laid off, namely, 42 inches; then each of these great divisions being subdivided into 11 equal parts, one of these parts will be equal to four teeth upon the radius of the wheel, so that the whole line CD will be divided into 88 radial pitches.* Next on the line CB set off the pitches which may be required in the scale, and through these points draw the 24 parallels to AD, terminating in the lines AB and DB. Then each parallel measured from the line BC to its point of termination in BD, is the radius of a wheel of 88 teeth, of the particular pitch marked against it on the line AB. They also express the radii of wheels having less than 88 teeth when measured only to the corresponding point in the line joining B, and the divisional on CD, against which the number of teeth is marked. Thus the radius of a wheel of 52 teeth and $1\frac{7}{8}$ inch pitch, is *r s* = $15\frac{7}{16}$ inches very nearly. (The true answer by the table, page 83, is 30.8724 ÷ 2 = 15.4362 inches.)

The scale may also be used when the number of teeth exceeds 88; for example, to find the radius of a wheel having 100 teeth. Thus having found the radius answering to 88 teeth; upon the same parallel take off the measure answering to the difference 100 — 88 = 12 teeth; and the two measures together will be the radius required.

To adapt the scale to odd numbers of teeth, the first division on the right of C is divided into single radial pitches, so that the radius of any wheel may be measured off without having recourse to calculation of any kind. Thus for example, if the wheel is intended to contain 50 teeth, the compasses being extended from 52 to the intersection of the parallel answering to the particular pitch to where it meets the line joining 2 and B, will give the radius required, that is, a radius answering to 52 — 2 = 50 teeth; and any other number of teeth when not marked against the base may be found in the same way, observing that it is more convenient to subtract than to add in this use of the scale.

For the proportions of the teeth set off C *a* = 7-*tenths* of the pitch, then will A*a* = 3-*tenths* of the pitch which corresponds to the depth from the point of the tooth to the pitch line. Again set off C*b* = 7-*fifteenths* of the 3 inch pitch, and 5-*elevenths* on the parallel against the 1 inch pitch; this will be the thickness of the tooth, allowing from a *fifteenth* for clearance on the largest pitch, to a *tenth* on those from $\frac{7}{8}$ inch and under; and A*b* will be the width of space, including the clearance. Lines being drawn from those points to B, complete the diagram, which will be found to contain all the proportions enumerated in the preceding table.

To use the scale lay off the addendum of the tooth, that is, the length beyond the pitch line, equal to A*a* = $\frac{3}{10}$ pitch, and the same length marked off within the pitch line will give the whole working depth of the tooth, namely, $\frac{6}{10}$ pitch. Then with the measure C *a* = $\frac{7}{10}$ pitch in the compasses, mark off the whole length of the tooth and this will allow $\frac{1}{10}$ at bottom for clearance. Again set off the thickness of tooth = C *b*, and the space = A *b* which will contain the clearance for the particular pitch, varying from $\frac{1}{15}$ to fully $\frac{1}{10}$ on the small pitches. It is hardly necessary to observe that these measurements must be taken upon the parallel corresponding to the particular pitch under consideration at the time.

The amount of bottom clearance is here presumed to be uniformly $\frac{1}{10}$ of the pitch; but if it be thought advisable to make this vary as in the case of the lateral clearance, it will then be necessary to insert a third line c B in the scale, and so related to a B, that the space $a\ c$ shall be throughout equal to the depth of tooth from the pitch circle to the root, and giving any bottom clearance that may be desired.

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* The mode of division here adopted, though sufficiently accurate for all practical purposes, is not arithmetically correct, each of the great divisions of seven times the pitch being too little by about the *six thousandth* part of an inch. Each radial pitch, it is obvious, for a pitch of 3 inches, ought to measure $\frac{3 \text{ inches.}}{3.1416 \times 2} = 0.4776$ inch, so that 44 of them ought to measure 0.4774 inch × 44 = 21.0056 inches, instead of 21 inches, which our mode of construction gives. The mode described is, however, easy in practice, and cannot lead to any sensible error; it is indeed the most certain, as no degree of care can ensure a nearer approximation to truth by actual measurement.

In relation to the strength of wheels, M. Morin in his *Aide Mémoire de Mécanique Pratique* gives it as a rule that when the velocity of the pitch circle does not exceed 5 feet per second, the breadth of the tooth measured parallel to the axis ought to be equal to 4 times the thickness; but when the velocity is higher, the breadth ought to be equal to 5 thicknesses, the teeth being constantly greased. If the teeth be constantly wet he recommends the breadth to be made equal to 6 thicknesses at all velocities.

With respect to the thickness of the tooth, it is plain that it must be dependent on the pressure which the tooth is required to sustain. This relation may be conveniently expressed for all cases by the formula

$$t = c\sqrt{\mathrm{W}}.$$

where t is the thickness of the tooth, W the pressure upon it in lbs. and c a constant multiplier depending upon the nature of the material of which the tooth is formed.

Now to find the value of c let us put a to represent the known strength of a bar 1 inch long, 1 inch thick, and 1 inch broad: then to support a weight W by a bar of a length l, and breadth b, the thickness t, must, according to the received formula for the strength of material, be in the relation

$$t = \sqrt{\frac{\mathrm{W} \times l}{a \times b}}.$$

But it has been shown that double the length is the only part of the breadth of the tooth which ought to be taken into the calculation; therefore we have $b = 2\ l$,

$$\text{whence } t = \sqrt{\frac{\mathrm{W} \times l}{a \times b}} = \sqrt{\frac{\mathrm{W} \times l}{a \times 2\ l}} = \sqrt{\frac{\mathrm{W}}{2a}}.$$

Now taking a=8000lbs. for cast iron, we have $2\ a$ = 16,000 lbs. But this is estimated as dead weight, and according to practice is consequently *ten* times greater than the working pressure, which such bar ought to be made to sustain. The true practical value of $2\ a$ may therefore be taken at 1600 lbs. and accordingly,

$$t = \sqrt{\frac{\mathrm{W}}{2\ a}} = \frac{\mathrm{W}}{1600} = \tfrac{1}{40}\sqrt{\mathrm{W}} = .025\sqrt{\mathrm{W}}.$$

Therefore for cast iron $c = .025$; and reasoning in the same way for brass, we find it $=.035$; for hardwood, $=.038$; so that for the thickness of teeth of these materials we have

$$\left.\begin{array}{ll} \text{For cast iron,} & t = .025\sqrt{\mathrm{W}} \\ \text{— brass,} & t = .035\sqrt{\mathrm{W}} \\ \text{— hardwood,} & t = .038\sqrt{\mathrm{W}} \end{array}\right\} \begin{array}{l} \text{which give } t \text{ in inches} \\ \text{or parts of an inch, W} \\ \text{being taken in lbs.} \end{array}$$

As an example of the application of these formulæ, let it be required to find the thickness of a tooth (cast iron,) which is to sustain a pressure of 4000 lbs. at the pitch circle, the breadth being double the length.

Here W = 4000; therefore $\sqrt{\mathrm{W}} = 63.25$.

Hence $t = .025\sqrt{\mathrm{W}} = .025 \times 63.25 = 1.58$ inch.

The same calculation applied with the formula for brass, would give $t = 2.21$ inches; and for wood, it gives $t = 2.4$ inches.

The thickness of tooth given by our rules is intended to make allowance for wear at a velocity of three feet per second, and has been found to be sufficient in practice. It is however, less by a small fraction, than would be found by application of Mr. Tredgold's rule,* which is deduced from the general expression $\mathrm{W}l = \frac{f.b.t}{6}$, where f is a constant depending upon the nature of the material, and b and d are respectively the breadth and depth (thickness) of the beam. This formula the author reduces, to make allowance for irregularity of action whereby the whole stress may be thrown, as already explained, upon one corner of the tooth, to $\mathrm{W} = \frac{f.\ t^2}{3}$ supposing the tooth to be everywhere of the same thickness: to make allowance for diminution of thickness he again puts $\mathrm{W} = \frac{f.\ t^2}{5}$. Finally, he makes allowance for wear at 3 feet per second, and this he observes will be amply enough at $\frac{1}{3}$ of the thickness; whence he obtains

$$\mathrm{W} = \frac{f.\ t^2 (1 - \frac{1}{3})^2}{5} = \frac{f.\ t^2}{10.25}.$$

For cast iron he takes $f = 15{,}000$, whence by substitution and reduction he obtains the rule,

$$\frac{\mathrm{W}}{1500} = t^2, \text{ or } \sqrt{\left\{\frac{\mathrm{W}}{1500}\right\}} = t.$$

That is, divide the stress at the pitch circle in lbs. by 1500, and the square root of the result is the thickness of the tooth in inches.

To compare this rule with that given above, let the pressure at the pitch circle be 4000 lbs. as before, then $\frac{4000}{1500} = 2\frac{2}{3}$; and the square root of $2\frac{2}{3}$ is 1.63 inch, which exceeds the answer previously found by the $\frac{1}{20}$th of an inch (that is, by $1.63 - 1.58 = .05$ inch). Either of these rules may therefore be employed indifferently, and, arithmetically considered, they admit of equally ready application. For the sake of comparison, we subjoin the following table, which will likewise be found useful in calculating the strength of wheels.

Stress in lbs. at the pitch circle.	Thickness of Teeth.		Actual pitches to which the wheels would be made.	Corresponding thickness, allowing 1-10th for clearance.
	By Author's Rule.	By Tredgold's Rule.		
lbs.	Inches.	Inches.	Inches.	Inches.
400	0.50	0.52	$1\frac{1}{8}$ to $1\frac{1}{4}$	0.536 to 0.593
800	0.71	0.75	$1\frac{1}{2}$ — $1\frac{5}{8}$	0.714 — 0.774
1200	0.87	0.90	$1\frac{7}{8}$ — 2	0.893 — 0.952
1600	1.00	1.03	2 — $2\frac{1}{8}$	0.952 — 1.012
2000	1.12	1.15	$2\frac{1}{4}$ — $2\frac{3}{8}$	1.031 — 1.132
2400	1.22	1.26	$2\frac{1}{2}$ — $2\frac{5}{8}$	1.190 — 1.250
2800	1.32	1.36	$2\frac{5}{8}$ — $2\frac{3}{4}$	1.250 — 1.309
3200	1.41	1.43	$2\frac{7}{8}$ — 3	1.369 — 1.429
3600	1.50	1.56	$3\frac{1}{8}$ — $3\frac{1}{4}$	1.488 — 1.548
4000	1.58	1.63	$3\frac{1}{4}$ — $3\frac{3}{8}$	1.548 — 1.607
4400	1.66	1.70	$3\frac{3}{8}$ — $3\frac{1}{2}$	1.607 — 1.667
4800	1.73	1.78	$3\frac{1}{2}$ — $3\frac{5}{8}$	1.667 — 1.726
5200	1.80	1.86	$3\frac{5}{8}$ — $3\frac{3}{4}$	1.726 — 1.786
5600	1.87	1.93	$3\frac{3}{4}$ — 4	1.786 — 1.904
6000	1.94	2.00	4 — $4\frac{1}{4}$	1.904 — 2.024

The last column of this table is calculated from the expression $pitch = t.(2 + \frac{1}{10}) = 2.1\ t.$ the clearance being a tenth. The thickness, allowing $\frac{1}{15}$ for clearance, may in like manner be calculated from the expression $pitch = t.\ (2 + \frac{1}{15}) = 2.067\ t$.

The formula reduced to expressions giving the pitch p in the same manner as the thicknesses are given above, will stand thus :—

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* Practical Essay on the strength of cast iron, p. 162, Fourth Edition, with Notes, by Eaton Hodgkinson, F.R.S. Weale, 1842.

Clearance a tenth. Clearance a fifteenth.

For cast iron teeth $p = .0525 \sqrt{W}$ and $p = .0517 \sqrt{W}$.
— brass teeth $p = .0735 \sqrt{W}$ and $p = .0723 \sqrt{W}$.
— wooden teeth $p = .0798 \sqrt{W}$ and $p = .0785 \sqrt{W}$.

By means of these rules the pitch may be directly calculated. Thus, for a pressure of 15,000 lbs. we have,

$$\sqrt{W} = \sqrt{15,000} = 122.5.$$

Consequently the pitch of a cast-iron wheel capable of sustaining that pressure at the pitch circle, allowing $\frac{1}{15}$ for clearance, will be

$$.0517 \sqrt{W} = .0517 \times 122.5 = 6.333 \text{ inches.}$$

The wheel from which this example is taken, has been several years in action, and has an actual pitch of 6 inches, and the mean pressure at the pitch circle is 14,786 lbs.

It may be here observed, that it is common, in calculations relative to the strength of the teeth of wheels, to make additional allowance for wear of the pinion; for if the pinion make double the number of revolutions made by the wheel with which it is engaged, its teeth will manifestly be subject to double the amount of wear by friction, consequently, to proportion the teeth of the pair so that they shall wear equally long, it is necessary to give an allowance of thickness on those of the pinion equivalent to the increase of abrasion to which they are subject. If we assume $\frac{1}{3}$ of the thickness as the proper allowance, this will give for the thickness of the pinion the expression $\frac{(2+n)\,t}{3}$, in which n is the number of revolutions which the pinion makes for one revolution of the wheel, and t the thickness of the teeth of the wheel. Thus for example, if the pinion make $2\frac{1}{2}$ times as many revolutions as the wheel, of which the thickness of the teeth is 1.12 inch; then the thickness of those of the pinion will be

$$\frac{t\,(2+n)}{3} = \frac{1.12\,(2+2.5)}{3} = 1.68 \text{ inch.}$$

This amount of difference is not however commendable in practice, at least in spur-geer, and it is therefore rarely adopted, as wheels would in that case require to be constantly made in pairs, which would lead to an endless accumulation of wheel patterns. Instead of making the allowance spoken of, the common practice is to adopt a larger pitch, rather indeed, to use wheels somewhat beyond the strength which is requisite for the work. The objection, it is true, does not apply to bevil-geer; still, the practice is that stated, and we see no very valid reason to relinquish it in favour of that to which the theory conducts, at least for the ordinary purposes of mill-work: particular cases may arise in which it may be desirable to adopt the more economical mode, as respects material, and were wheels made of the precious metals, we would undoubtedly recommend it for general practice; but so long as the very best cast-iron wheels can be purchased for 12s. a cwt., the little increase of weight which is requisite to accomplish the same end, and in general more effectually, is well and economically bestowed.

It may be also observed that our rule gives the stress, which the individual teeth of a wheel are capable of sustaining, and presumes that of the pair of wheels engaged only two teeth are in contact. This condition will hold when the difference of size of the wheel and pinion and the form of the teeth are such as to produce contact only on the line of centres: if the pinion be very small, compared with the wheel, the contact at the pitch circle will be correspondingly reduced in extent, and as the diameters approximate to equality, the number of teeth in contact will be proportionally augmented. Were we, therefore, to adhere strictly to the circumstances which theory indicates, it would be necessary, before pronouncing on the power which any given pair of wheels are sufficient to bear, to ascertain the number of teeth over which the working pressure would be distributed; and this being determined, the strength of the pair will be as many times greater than that indicated in the table, as there are teeth of each wheel in contact. Thus, if the sizes of the pair be such as to bring two teeth constantly into action, then the strength will be double that given by the table; if the contact extend to three teeth, the strength will be three times greater than the pressure given.

In making such allowance, it is, however, presumed that the wheels work with mathematical accuracy; but it is to be kept in mind that in practice a very slight inaccuracy—even the interposition of any hard body, however small, and more especially, any inequality, or irregularity, on the side of one of the teeth, throws the whole strain upon a single tooth; consequently, if the teeth be not made of sufficient strength to sustain individually the whole pressure to which the wheel is liable, should such casualty occur, or should such inequality happen through imperfect workmanship, irregularity of cooling of the wheel when cast (supposing it to be of metal), the wheel would be *stripped* of several of its teeth, an accident which not unfrequently occurs, even under the most favourable circumstances. For these reasons, it is advisable, in practice, to consider the strength of a single tooth as representing the strength of the wheel, and for reasons formerly stated (page 88), the strength of the tooth ought to be taken under circumstances the least effective, namely, when the pressure acts exclusively on one of its corners. Besides simplifying greatly the conditions of the question, this is manifestly the safest practice, and we may add, it is that adopted when any other than the *rule of thumb* is followed.

It has already been shown that by a horse power is meant a pressure of 33,000lbs., moved with a velocity of *one* foot in a minute, equivalent to 550lbs. raised 1 foot in a second. But this is the mean of the force exerted, and as most prime movers are more or less variable in their motion, any wheel required to transmit that motion should be strong enough to bear the maximum force with safety. The excess of the maximum over the mean force ought of course to be specially considered in those cases where its amount is particularly remarkable; but for ordinary and general purposes we may assume, as a very safe approximation, that it exceeds the mean of the whole force exerted by the fraction $\frac{3}{11}$. Making that allowance, we shall have, as the practical strain of a horse power, 550lbs. $\times\ 1\frac{3}{11} =$ 700lbs. raised 1 foot per second. By substitution of this value of the horse power, in the rule formerly given (page 87), it will become

$$\frac{700}{v} \times H = \text{the stress on the teeth of the wheel in lbs.}$$

Having determined from this rule the pressure at the pitch circle of a wheel, the requisite thickness of tooth may be calculated by application of the rule for finding the thickness.

As an example, let a steam engine of 12 horse power be applied to drive the machinery of a factory, and let it be required to find the strength of the teeth of a first wheel on the main shaft, which will have a velocity of 4 feet a second, at the pitch circle?

Here $H = 12$ and $v = 4$; therefore $\frac{700}{v} \times H = \frac{700}{4} \times 12 = 2100$lbs., the pressure at the pitch circle $= W$ in the rule for the thickness of teeth.

Now the square root of 2100 is 45.826, and supposing the wheel to be of cast-iron, then we have

$$t = .025\sqrt{W} = .025 \times 45.826 = 1.1456 \text{ inch},$$

the thickness of the teeth of the wheel; consequently, if $\frac{1}{10}$ be allowed for clearance, the pitch will be $1.1456 \times 2.1 = 2.52$ inches. The actual pitch would therefore be made from $2\frac{1}{2}$ to $2\frac{5}{8}$ inches.

If the wheel have wooden teeth, then the rule $t = 038\sqrt{W}$ must be used, which gives 1.74 inch as the thickness of tooth, and 3.66 inches as the pitch; the pitch to be adopted would therefore be $3\frac{3}{4}$ inches.

On the subject of wooden teeth, it may not be out of place to remark, that their strength is not unfrequently underrated. In Robertson Buchanan's Essays on Mill Work, it is stated that "the kind of wood employed for teeth is usually $\frac{1}{4}$th of the strength of cast-iron; and since the thickness of the teeth should vary inversely as the square root of the power (strength) of the material, the square root of $\frac{1}{4}$ being $\frac{1}{2}$, wooden teeth should be twice the thickness of cast-iron teeth. The pitch of course will vary in the same proportions."

This statement is very far from being verified in practice. The kind of workmanship, indeed, which the author found most common at the time he wrote, in some measure bears him out; but when finished according to the invariable mode now adopted by millwrights of character, the rule which we have given above makes full allowance for the difference of strength of the two materials; and even forty years ago, examples were not wanting of mortise-wheels of the highest degree of finish. Those of the Royal Mint, constructed by the late Sir John Rennie, which are stated to be, in every respect, as efficient at the present day as they were on making their first revolution, afford good evidence of the durability of wooden teeth, and also of their strength; for in the operation of coining, the strain is not of that degree of uniformity best suited to the equal wear of the geering. When cast-iron and wooden teeth work together, their action upon each other tends, in consequence of the elasticity of the wood, to maintain a more uniform distribution of the strain, and being at first commonly more accurately dressed, to prevent abrasion of the wood by the iron, they work with much less friction, are less liable to shocks, and nearly exempt from accident by hard particles coming between the teeth.

The best practice, when a mortise and iron wheel are to work together, is to make both of the same pitch, and in the first instance, of the same thickness of tooth—the pitch being of course calculated for the wooden wheel by the rule $p = .0785\sqrt{W}$, which allows $\frac{1}{15}$ for clearance, (and with good workmanship this is amply sufficient in cases of the kind proposed); afterwards to dress down the teeth of the iron wheel, by the *chipping* tool, or the wheel-cutting machine, and file, to the exact form and thickness, as given by the formula $t = .025\sqrt{W}$; that is, to a thickness in relation to the thickness of the wooden teeth, which shall have the ratio of 25 to 38. Both of the wheels will then be of the same strength, and from their superior finish, will work with much less friction, and, consequently, less wear than if both wheels were of iron.

It was stated (page 87), that the stress on the teeth of a wheel may be represented by $\frac{P \times V}{v}$ where P is the power of the prime mover in lbs., V the velocity of that power in feet per second, and v the velocity at the pitch circle of the wheel. Now if m be the number of revolutions to be made by the wheel in a minute, and n the number of teeth to be cut on it, p the pitch in *parts of a foot*, and W the pressure upon each tooth in lbs.; then np will represent the circumference of the pitch circle, and $mnp = v$ the space in feet described by it per minute. But P represents the work transmitted by the wheel through the space mnp, therefore $\frac{P}{mnp} = \frac{P}{v}$ represents the *mean* pressure under which that work is done $= W$. Now the pitch of the teeth is expressed, as has been shown, by the formula $p = c\sqrt{W}$, where c is the representative of the multipliers given in the preceding rules; consequently, by eliminating W between this and the expression found above, we obtain

$$p = \sqrt[3]{\frac{c^2 P}{mn}},$$

which is an expression from which the pitch might be readily calculated. But the operation may be put in a still simpler form, by reducing P to the received unit of a horse power. For the *mean* pressure on the teeth, P would be represented by 33,000 H; but making the same allowance as before for variableness of motion we have $P = 42,000$ H. Accordingly by substituting this value in the preceding equation, and representing the resulting constant, $42,000\, c^2$ by C^3, we obtain

$$p = C\sqrt[3]{\frac{H}{m\,n}} \text{ in parts of a foot.}$$

And the numerical values of C being deduced for the several materials and amount of clearance, ($\frac{1}{10}$th of pitch,) we shall have the following values of the pitch in inches and parts of an inch, agreeable to the method adopted in the preceding rules, namely,

for cast-iron $p = 12\sqrt[3]{\frac{H}{mn}}$, or $\sqrt[3]{\left\{\frac{1728\ H}{m\,n}\right\}} = p$.

for brass, $p = 13\sqrt[3]{\frac{H}{mn}}$. or $\sqrt[3]{\left\{\frac{2197\ H}{m\,n}\right\}} = p$.

for hardwood. $p = 14\sqrt[3]{\frac{H}{mn}}$, or $\sqrt[3]{\left\{\frac{2744\ H}{m\,n}\right\}} = p$.

Conversely, the power which the teeth of a wheel of given pitch are capable of transmitting may be readily calculated from the following rules, which are immediately deduced from the above,

For cast-iron, $H = \frac{m\,n\,p^3}{12^3}$ or $\frac{m\,n\,p^3}{1728}$.

For brass, $H = \frac{m\,n\,p^3}{13^3}$ or $\frac{m\,n\,p^3}{2197}$.

For hardwood, $H = \frac{m\,n\,p^3}{14^3}$ or $\frac{m\,n\,p^3}{2744}$.

Thus supposing the pitch $p = 3$ inches, the number of teeth $n = 60$, and the number of revolutions n which the wheel is designed to make per minute $= 50$; then the wheel being cast-iron, the power H which it is capable of transmitting will be

$$H = \frac{60 \times 50 \times 3^3}{1728} = 47 \text{ horses' power nearly.}$$

This wheel is employed to transmit 45 horses' power at the given velocity, and has been at work for several years.

As another practical test of the value of these rules, we shall apply the first to the case of a cast-iron wheel in actual use, and which seems to be very nearly balanced in strength with the work it is required to perform, the teeth having failed twice without apparent cause. The power transmitted by the wheel is that of a steam-engine, rather two steam-engines, the *mean* power of which, by the indicator, is a very little above 140 horses' power; the machinery driven is that of a cotton factory, and may be considered regular; the number of turns m, of the wheel per minute is 63, number of teeth $n = 72$; the lateral clearance of the teeth is about a *twelfth*, so that for the pitch p, we ought to take a co-efficient, slightly less than 12; but taking the mean we have,

$$p = 12\sqrt[3]{\frac{H}{mn}}.$$

And substituting in this the numerical values stated, we have

$$p = 12\sqrt[3]{\frac{140}{63 \times 72}} = 12 \times .31 = 3.62,$$

which is the pitch in inches the wheel ought to have according to our rule, that it may be sufficient for the work it has to perform. Now the actual pitch is 3.25 inches, that is, nearly $\frac{37}{100}$ inch less than the rule gives. Our rule, moreover, is founded on the principle that additional strength is not attained by extending the breadth of the teeth beyond double the length; according to this principle, the wheel in question would have the same strength at five inches breadth that it has with a breadth of 12 inches. For the reasons previously assigned, the latter breadth, however, has important advantages.*

Tredgold, in his treatise on cast-iron, expounds an analogous rule, which is deduced in the manner following. Observing that we cannot always know the velocity of the pitch circle of a wheel, because it is not in general possible to vary the number of teeth after the pitch is determined, so as to give it the velocity assigned to it before the pitch was known, he proceeds to deduce a formula for the thickness of the teeth thus: Let N be the number of revolutions the axis is to make per minute, on which the wheel is to be placed; and R the radius the wheel should have if the pitch were 2 inches, then the velocity v of the pitch circle will be expressed thus.

$$v = \frac{2.1\, t \times N \times R}{19.09 \times 24} = \frac{t\, N\, R}{218.16}.$$

Consequently P being the power of the first mover in lbs. and V the velocity of that power in feet per second, we have

$$\frac{P\,V}{v} = \frac{218.16\, P\, V}{t\, N\, R} = W, \text{ the stress on the teeth.}$$

But as was formerly shown, the thickness t of the teeth is expressed by $\frac{W}{1500} = t^2$ or $\sqrt{\frac{W}{1500}} = t$; hence by substitution there is obtained

$$\frac{218.16\, P\, V}{1500\, N\, R} = t^3; \text{ or } \sqrt[3]{\left\{\frac{.14544\, P\, V}{N\, R}\right\}} = t;$$

which expressed in words is the following rule: Multiply 0.146 times the power of the first mover in lbs. by its velocity in feet per second, and divide the product by the number of revolutions the wheel is proposed to make per minute, and by the radius the wheel should have if its pitch were 2 inches; the cube root of the quotient will be the thickness of the teeth in inches.

This rule is illustrated by the two following examples: 1st. Suppose the effective force acting at the circumference of a water-wheel to be 300lbs. and its velocity 10 feet per second, it is proposed to find the thickness for the teeth of a wheel which is to make 12 revolutions per minute, and to have 30 teeth.

Here $0.146 \times 300 \times 10 = 438$.

And since the radius of a wheel with 30 teeth and a pitch of 2 inches is 9.567, (as may be ascertained by the rule p. 83,) we have

$$t^3 = \frac{438}{12 \times 9.567} = 3.815.$$

and $\therefore t = \sqrt[3]{3.815} = 1.563$ nearly, the thickness of the teeth required in inches, (and the pitch will therefore be $1.563 \times 2.1 = 3.2823$ inches.) The wheel must be understood to be of cast-iron.

To compare our rule with this, we shall apply it to the same example. Our rule expressed in words will be this: Divide the number of horses' power of the prime mover (estimated at 33,000 lbs. raised one foot per minute), by the number of revolutions which the wheel is to make per minute and by the number of teeth which the wheel is to have; extract the cube root of the quotient and multiply the result by 12 for cast-iron wheels with a clearance of $\frac{1}{10}$th pitch. For wheels of brass the multiplier is 13, and for hardwood teeth 14. The final product in each case is the pitch in inches and decimals of an inch.

Now in the example given, the number of horses' power is $H = (300 \text{ lbs.} \times 10 \times 60) \div 33{,}000 = 5.45$; the wheel is to make $m = 12$ revolutions a minute, and the number of teeth $n = 30$; hence we have $\frac{H}{mn} = \frac{5.45}{12 \times 30} = .01514$.

And the cube root of .01514 is .25 very nearly; and this multiplied by 12 gives $.25 \times 12 = 3$ inches as the pitch of the teeth, a result less by .28 inch than that found by Tredgold's rule.

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* The rules here given are founded on the pressure which the several materials are capable of sustaining without rupture; but when the motion is very irregular, it has been found that a brass wheel is capable of sustaining more pressure than a cast-iron one of equal pitch. This seems accounted for by the greater ductility of the metal which allows of a certain amount of yield in cases where the action is so irregular as to cause the teeth to strike upon one another, and which, therefore, brings into operation another property of the material, namely, their capability of resisting percussion. The same remarks apply in some measure to wheels having hardwood teeth, which, instead of being weaker, are under certain circumstances stronger than cast-iron teeth of the same pitch.

The following is Mr. Tredgold's second example.—Let the effective force on the piston of a steam-engine be 6875 lbs. and its velocity $3\frac{1}{2}$ feet per second; it is required to determine the strength of the teeth of a wheel to be driven by this engine, which is to have 152 teeth and make $17\frac{1}{2}$ revolutions per minute.

In this case the radius for 152 teeth with a 2 inches pitch is 48.387 inches.

$$\text{therefore } \frac{0.146 \times 6875 \times 3.5}{17.5 \times 48.387} = 4.15.$$

And the cube root of 4.15 is very nearly 1.6 inch the thickness of the teeth; the pitch is therefore $1.6 \times 2.1 = 3.36$ inches.

To apply our rule to this example we must find the value of H from the data given. Now $3\frac{1}{2}$ feet per second is 210 feet per minute, and the pressure on the piston is 6875 lbs; therefore

$$\text{H} = \frac{210 \times 6875}{33000} = 44 \text{ nearly.}$$

Hence, taking the second form of our rule, which is in some respects more convenient, we have

$$\frac{1728 \text{ H}}{m\, n} = \frac{1728 \times 44}{17.5 \times 152} = 28.553$$

which is the third power of the pitch; therefore

$$p = \sqrt[3]{28.553} = 3.057 \text{ inches.}$$

and which is $\frac{3}{10}$ of an inch less than the pitch by Mr. Tredgold's rule.

Robertson Buchanan, in his article on wheels, has another form of rule, which is in some respects very simple. It is thus expressed for the thickness,

$$t = \tfrac{3}{4}\sqrt{\frac{\text{H}}{v}};$$

in which the letters have the same significations as before. In words it is this: Find the number of horses' power which are equivalent to that of the first mover of the train of machinery, and divide that number by the velocity in feet per second, of the pitch line of the wheel; extract the square root of the quotient, and three-fourths of this root will be the least thickness of the tooth in inches.

Applying this rule to the last example given, we have H $=44$, and taking the pitch at 3 inches, the velocity, at the pitch line of the wheel, will be $11\frac{1}{12}$ feet per second.

$$\text{Hence } t = \tfrac{3}{4}\sqrt{\frac{\text{H}}{v}} = \tfrac{3}{4}\sqrt{\frac{44}{11\frac{1}{12}}} = \tfrac{3}{4} \times 1.989 = 1.492,$$

the thickness in inches; therefore the pitch is 3.133 inches, which exceeds the pitch assumed by $\frac{1}{8}$th of an inch, and is very nearly intermediate to our own and Mr. Tredgold's rule.

Every writer on the teeth of wheels has thought it necessary to adduce rules for finding the proper breadth of the teeth. As respects strength, such a calculation has been shown to be in a manner unnecessary beyond merely doubling the length, which is immediately deducible from the pitch; and as respects durability there seems to be no theoretical limit to the breadth, for the more the pressure and rubbing action is diffused, the less rapidly will the teeth be worn. The breadth to be assigned in practice must, therefore, be always a quantity to be determined by circumstances, and modified by the particular opinions of those concerned; if the motion be particularly uniform and free of vibration, the breadth may be extended even to four times the pitch with advantage; but if the contrary circumstances obtain, this great breadth will in like manner have a contrary effect, the teeth becoming frequently locked together, will more speedily wear out of shape if they be made too strong to twist and break one another out at the ends. When the shafting is light, this is a frequent occurrence, although the reason does not seem to be always understood; and accordingly, the remark is not uncommon, that the wheel ought on account of its great breadth to have been more than sufficient for the work. However contradictory it may appear, strength in this sense is not an unfrequent cause of weakness and failure.

But as these notions are opposed to the general opinion and practice of millwrights, it may be necessary, to save ourselves from the charge of improper omission, to exhibit briefly the *principles* upon which the question of breadth is commonly determined.

The rule propounded by Robertson Buchanan, in his Essays on Mill Work, is a very simple one. Observing that three inches is a very common pitch for first motions of mills, he directs that, for this pitch, and a velocity of three feet per second at the pitch line, in situations where the wheels cannot be properly and constantly greased, the teeth shall be as many inches broad, as the number of horses' power which they are required to transmit. And in common practice, for a pitch of 3 inches and the given velocity, every inch of breadth may be valued at one and a half horse's power. Thus, for example, *nine horses' power* will require a wheel of *six inches broad* for its transmission, when the velocity is 3 feet a second. This is the rule adopted by Tredgold, who illustrates it more fully by assuming a horse power at 200lbs., with a velocity of 3 feet a second: the breadth in inches will then, he observes, be equal to the horses' power, when the velocity of the pitch circle is $1\frac{1}{2}$ foot per second; twice the breadth will be the horses' power, when the velocity is 3 feet per second; three times the breadth will be the horses' power, when the velocity is $4\frac{1}{2}$ feet a second; four times the breadth will be the horses' power, when the velocity is 6 feet per second; and generally n times the horses' power, when the velocity of the pitch circle is n times $1\frac{1}{2}$ foot per second.

Messrs. Roberton and Carmichael's rules, which are still implicitly followed in several engineering establishments, are exhibited in the following tables which are thus constructed. The *thickness* is reduced at each step $\frac{1}{10}$th of an inch; the *breadth* is taken at four times the thickness; and the pitch is found by multiplying the thickness by 2.1. According to Roberton's rule, the *strength* of the teeth is ascertained by multiplying the square of their thickness into their breadth taken in inches and parts of an inch: the number that represents the strength, also represents the number of horses' power, at a velocity of about 4 feet per second. Thus, where the pitch is 3.15 inches, the thickness 1.5 inch, and the breadth 6 inches, the strength is valued at $13\frac{1}{2}$ horses' power, with a velocity of 4 feet per second at the pitch line.

Carmichael's rule is this: Multiply the breadth of the teeth by the square of the thickness, and divide the product by the length; the quotient will be the proportionate strength in horses' power, with a velocity of 2.27 feet per second.

The results of these rules, from 4-inch to 1-inch pitch, are given in the following tables:—

MR. ROBERTON'S TABLE.

Pitch in Inches.	Thickness of Teeth in Inches.	Breadth of Teeth in Inches.	H. P. at 4 feet in 1″.	H. P. at 3 feet. in 1″.	H. P. at 6 feet in 1″.	H. P. at 8 feet in 1″.
3.99	1.9	7.6	27.43	20.57	41.14	54.85
3.78	1.8	7.2	23.32	17.49	34.98	46.64
3.57	1.7	6.8	19.65	14.73	29.46	39.28
3.36	1.6	6.4	16.38	12.28	24.56	32.74
3.15	1.5	6.	13.5	10.12	20.24	26.98
2.94	1.4	5.6	10.97	8.22	16.44	21.92
2.73	1.3	5.2	8.78	6.58	13.16	17.54
2.52	1.2	4.8	6.91	5.18	10.36	13.81
2.31	1.1	4.4	5.32	3.99	7.98	10.64
2.1	1.0	4.	4.0	3.0	6.0	8.0
1.89	.9	3.6	2.91	2.18	4.36	5.81
1.68	.8	3.2	2.04	1.53	3.06	3.08
1.47	.7	2.8	1.37	1.027	2.04	2.72
1.26	.6	2.4	.86	.64	1.38	1.84
1.05	.5	2.	.5	.375	.75	1.

MR. CARMICHAEL'S TABLE.

Pitch in Inches.	Thickness of Teeth in Inches.	Breadth of Teeth in Inches.	Length of Teeth in Inches.	H. P. at 2.27 feet in 1″.	H. P. at 3 feet in 1′.	H. P. at 6 feet in 1″.	H. P. at 11 feet in 1″.
3.99	1.9	7.6	2.28	12.03	15.90	31.80	58.30
3.78	1.8	7.2	2.16	10.80	14.27	28.54	52.32
3.57	1.7	6.8	2.04	9.63	12.72	25.54	46.68
3.36	1.6	6.4	1.92	8.53	11.27	22.54	41.32
3.15	1.5	6.0	1.80	7.50	9.91	19.82	36.33
2.94	1.4	5.6	1.68	6.53	8.63	17.26	31.64
2.73	1.3	5.2	1.56	5.63	7.44	14.88	27.28
2.52	1.2	4.8	1.44	4.80	6.34	12.68	23.24
2.31	1.1	4.4	1.32	4.03	5.32	10.64	19.54
2.10	1.0	4.0	1.20	3.33	4.40	8.81	16.15
1.89	0.9	3.6	1.08	2.70	3.57	7.14	13·09
1.68	0.8	3.2	0.96	2.13	2.81	5.62	10.33
1.47	0.7	2.8	0.84	1.63	2.15	4.30	7.88
1.26	0.6	2.4	0.72	1.20	1.59	3.18	5.83
1.05	0.5	2.0	0.60	0.83	1.10	2.20	4.03

Assuming that the teeth of wheels follow the same law of strength by increase of breadth as in thickness, and referring back to the general formula for the strength of a beam of given dimensions, we arrive at the conclusion that the breadth b and the length l are in the relation of $b = \frac{6}{5} l$; and supposing with some engineers of experience that a breadth of 6 inches is sufficient for a power equivalent to 9 horses, when the pitch line moves with a velocity of 3 feet per second, then we have the simple analogy

$$\frac{9 \text{ HP}}{3} : 6 \text{ inches} :: \frac{\text{H}}{v} : b\text{; whence } b = \frac{2\text{ H}}{v}:$$

That is the following rule: Double the number of horses' power of the prime mover, and divide the result by the velocity of the pitch circle in feet per second: the result is the breadth of the teeth in inches.

Thus the power transmitted by a wheel moving at $5\frac{1}{2}$ feet velocity per second at the pitch circumference, is 16 horses; the breadth of the wheel will therefore be

$$\frac{2 \times 16}{5\frac{1}{2}} = \frac{3.2}{5.5} = 5.82 \text{ inches nearly.}$$

Again, on the same principle, having the breadth and velocity of teeth given, the number of horses' power, which the wheel can transmit, is determined by the rule $\text{H} = \frac{v\,b}{2}$.

That is: Multiply the velocity of the pitch circle in feet per second by the breadth of the teeth in inches, and half the product will be the number of horses' power which the wheel is capable of transmitting.

Thus, let the breadth be 12 inches, and the velocity 21 feet per second; then 12 × 21 = 252, half of which is 126, the number of horses' power required.

It is easy to see, however, that unless the breadth be a function of the pitch, any calculation of this kind cannot be satisfactory; and from the remarks we have already made, there cannot exist much difficulty in fixing upon the proper breadth the wheel ought to have for the particular purpose intended.

The following table may be useful in determining the relation of the dimensions of the teeth of wheels of the given pitches, and the power which they are capable of transmitting safely at the various speeds named. The table was originally constructed by the writer from the formula

$$\frac{2\,p^2 \times v}{5} = \text{H},$$

and has since been extensively used.

Pitch.	Thickness of Teeth.	Length of Teeth.	Least breadth of Teeth.	Velocity of the Wheel at the pitch circle.				
				3 feet per second.	4 feet per second.	5 feet per second.	7 feet per second.	11 feet per second.
Inches.	Inches.	Inches.	Inches.	H. P.	H. P.	H. P.	H. P.	H. P.
6	2.9	4.2	8.4	$43\frac{1}{5}$	$57\frac{3}{5}$	72	$100\frac{4}{5}$	$158\frac{2}{5}$
$5\frac{1}{2}$	2.6	3.85	7.7	$36\frac{3}{10}$	$48\frac{2}{5}$	$60\frac{1}{2}$	$84\frac{7}{10}$	$133\frac{1}{10}$
4	1.9	2.8	5.6	19	$25\frac{1}{2}$	32	45	$70\frac{1}{2}$
$3\frac{1}{2}$	1.6	2.45	4.9	$14\frac{3}{4}$	$19\frac{1}{2}$	$24\frac{1}{2}$	$34\frac{1}{4}$	54
3	1.4	2.1	4.2	11	$14\frac{1}{2}$	18	25	$39\frac{1}{2}$
$2\frac{1}{2}$	1.2	1.75	3.5	$7\frac{1}{2}$	10	$12\frac{1}{2}$	$17\frac{1}{2}$	$27\frac{1}{2}$
2	0.95	1.4	2.8	$4\frac{3}{4}$	$6\frac{1}{4}$	8	11	$17\frac{1}{4}$
$1\frac{3}{4}$	0.83	1.225	2.45	$3\frac{1}{2}$	5	$6\frac{1}{4}$	$8\frac{1}{2}$	$13\frac{1}{2}$
$1\frac{1}{2}$	0.71	1.05	2.1	$2\frac{3}{4}$	$3\frac{1}{2}$	$4\frac{1}{2}$	$6\frac{1}{4}$	10
$1\frac{1}{4}$	0.59	0.875	1.75	2	$2\frac{1}{2}$	$3\frac{1}{8}$	4_{5}	$6\frac{4}{5}$
$1\frac{1}{8}$	0.53	0.7875	1.575	$1\frac{1}{2}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$3\frac{1}{2}$	$5\frac{1}{2}$
1	0.48	0.7	1.4	$1\frac{1}{5}$	$1\frac{3}{5}$	2	2^{4}	$4\frac{2}{5}$
$\frac{7}{8}$	0.41	0.6125	1.225	1	$1\frac{2}{5}$	$1\frac{3}{4}$	$2\frac{1}{2}$	$3\frac{4}{5}$
$\frac{3}{4}$	0.36	0.525	1.05	$\frac{7}{10}$	$\frac{9}{10}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$2\frac{1}{2}$
$\frac{5}{8}$	0.33	0.4375	0.875	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{7}{10}$
$\frac{1}{2}$	0.24	0.35	0.7	$\frac{3}{10}$	$\frac{2}{5}$	$\frac{1}{2}$	$\frac{7}{10}$	$1\frac{1}{10}$
$\frac{3}{8}$	0.18	0.2625	0.525	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{3}{10}$	$\frac{2}{5}$	$\frac{3}{5}$
$\frac{1}{4}$	0.12	0.175	0.35	$\frac{3}{40}$	$\frac{1}{10}$	$\frac{1}{8}$	$\frac{7}{40}$	$\frac{11}{40}$

To find the power which a wheel is capable of transmitting for other velocities than those in the table.—For 6 feet per second, double the result given for 3 feet; for 8 feet double the result at 4 feet, and so on; and for lower velocities than those given, divide the tabular number by the ratio which they bear to those enumerated. Thus, for $2\frac{1}{2}$ feet velocity, take half the result at 5 feet, and so of other velocities.

When a wheel and pinion, which differ very much in size, work together, the teeth of the latter, on account of their unequal thickness, are capable of sustaining much less pressure than the teeth of the wheel: they are in effect, if not in fact, much reduced in thickness. The reason of this, and the extent of the difference, and defect of strength, will appear in treating of the form of the teeth of wheels; but in the meantime, it is necessary to remark, in connection with our present subject, that in applying rules to the calculation of the strength of wheels, the difference of size of the pair ought not to be overlooked, unless, as is indeed very common in practice, the deficiency of strength be made up to the pinion by a flange cast on one or both sides of the rim, of the same depth as the teeth, and binding these together like the staves of a trundle. In this case the pinion is commonly the stronger wheel of the pair.

We conclude this part of the subject with the following table of the dimensions of cast-iron and mortise wheels in actual use, with the power they are employed to transmit.

TABLE OF THE DIMENSIONS OF WHEELS IN ACTUAL USE.

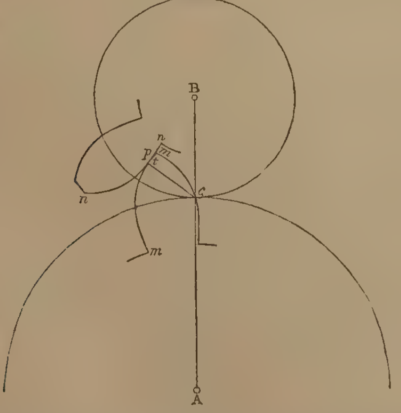
Pitch in inches.	Character of Wheel.	Number of teeth.	Breadth in inches.	No. of revolutions per minute.	Horses' power.	
					Actual.	Calculated.
4	Spur	38	15	80	105	·112
3	—	93	10	12	18	17
3	—	40	10	65	38	40
3	Fly wheel	208	12	40	110	130
2	Spur	104	5	25	12	12
2	—	12	6	87	3	5
1¾	Internal	135	5	12	2	1.6
1¾	Spur wheel	76	5	101	8	7.75
1½	—	72	4½	120	8	7.5
2¼	—	95	6	25	1½	1.676
3¼	Bevil wheel	40	7	30½	20	24.34
2⅝	Cog wheel	60	6	40	12	15.82
5½	Bevil wheel	70	12	10	60	67.396
2½	Spur wheel	90	8	12	6	9.72
3¾	Internal	80	9	20	41	48.8
3	Cog spur wheel	60	8	30	121	177
6	Spur wheel	30	14	7	21	26¼
4	—	100	10	8	25	29.6
2⅞	—	33	7	55	23	25
2¾	—	108	7	20	25	26
2½	Internal	100	7	10	87	90.4
5	—	60	12	12	55	53.5
5	Spur	41	10	20	61	59
4.76	—	50	12	23	65	71.3
3¾	Bevil wheel	35	10	24	26	25.6
4	Cog bevil wheel	50	10	28	33	32.6
4	Cog spur wheel	35	9	20	18	16.3
6	On water wheel	112	14	12	110	168
4¾	Spur wheel	55	10	16	56	54.56

SECTION V.—ON THE PRINCIPLES WHICH DETERMINE THE TRUE FORM OF THE TEETH OF WHEELS.

In the construction of wheels, the problem which presents itself relative to the shape of the teeth is this, that the surfaces of mutual contact shall be so formed that the wheels shall be made to turn by the intervention of the teeth, precisely as they would by the friction of their circumferences. This problem has, from its importance, engaged the attention of several very eminent mathematicians in this country, and on the continent of Europe; yet it cannot be concealed that the question has been as often complexed as elucidated by the multiplicity of mathematical refinements in which it has been involved. Without attempting to abandon the received methods of investigation, we shall endeavour to examine and illustrate the subject more as a question which concerns the mechanic than the mathematician—borrowing from the science of the latter those principles which it is the business of the mechanic to apply and reduce to practice.

That the problem is possible, may be shown by a very simple construction. Let *mm* and *nn* be the acting faces of two teeth of a pair of wheels of which that marked A is the driver; then it is plain that the faces of the teeth may be of such curvature in relation to each other, that as the circle A revolves and carries with it the circle B by their mutual contact at C, the teeth may continually touch one another throughout their lines of curvature, continually altering their relative positions and their point of contact *t*, as the primitive circles change their point of contact; and this being true, it evidently follows that the two circles would be made to revolve by the contact of teeth, whose surfaces are thus formed, precisely as they would by the friction of their circumferences at the point *c*. For observing the action of the circles upon each other, in the former case we find a series of points of contact passing through *c*, and bringing about a corresponding rotation of points of contact of the teeth at *t*; and in the latter case, the action being transferred to the teeth, the same transposition of the points of contact at *t* is followed by the same transposition of points, as formerly, through *c*.

Fig. 31.

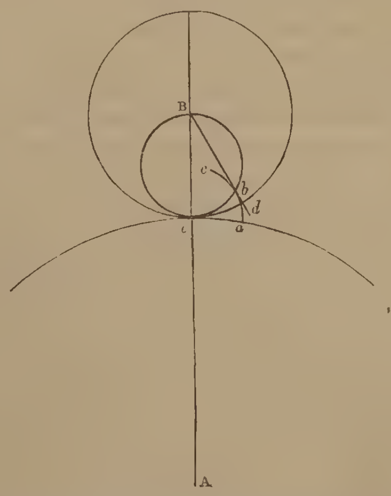


To form teeth whose surfaces of contact shall possess the property here assigned to the curves of the teeth *mm* and *nn*, is the problem to be solved; and the solution is dependent on the following

Fundamental principle.—In order that two circles A and B may be made to revolve by the contact of the surfaces of the curves *mm* and *nn* of their teeth precisely as they would by the friction of their circumferences, it is necessary and sufficient that a line drawn from the point of contact *t* of the teeth to the point of contact *c* of the circumferences (pitch circles), should in every position of the point *t* be perpendicular to the surfaces of contact at that point; that is, in the language of mathematicians, that the straight line be a *normal* to both the curves *mm* and *nn*.

In proceeding to establish this principle, we must in the first place recur to a proposition, which in effect is that employed by the mechanic in the use of his *templets* for drawing in the curve of the teeth. In fig. 32, let A and B be the centres of motion of two circles whose circumferences meet in the point *c*, and let *c*B*c* be a third circle in the same plane, and having its diameter equal to radius B*c*, and its point of contingence also at *c*. If this minor circle be made to revolve on its axis through an arc *cb*, and it thereby communicate an equal motion to the other two circles, moving them by the point of continual contact *c*, common to the three

Fig. 32.



circumferences, it is evident that all the parts of the arc *cb* of the impelling circle will have been applied in succession to all the parts of the equal arcs *ca* and *cd* of the primary circles, in exactly the same manner as if these last had remained immovable, and the minor circle *c*B*c* had been set free from its centre and rolled on their circumferences, exteriorly on A, and interiorly on B. If, then, a tracing point be fixed in the circumference of this circle, and all the axes be fixed in position, in passing from *c* to *b*, it will trace upon the plane of the circle A, the curve *ab* exterior to the circumference, and on the plane of the circle B, the straight line *db* interior to the circumference. This line will be a tangent to the curve at *b*, and, if produced, will meet the circumference of the tracing circle in B, the centre of the circle in which it revolves. This condition follows immediately from the relation of the diameters of the two circles; for these determine the relation of the circumferences, and therefore the relation of any equal arcs measured upon them. Now, the internal circle having as its diameter the radius of that within which it is supposed to revolve, will also have half the length of circumference of that circle; and, therefore, any portion *cb* of the less circumference will contain double the number of degrees of any portion *cd* of equal length measured upon the larger circumference. But the angle *c*B*b* being an angle at the circumference of its circle, and standing upon an arc of the *same* length as the angle *c*B*d*, at the centre of a circle of double the diameter of the other, and the side *c*B being common to the two angles, the adjacent side B*b* of the angle *c*B*d* will fall upon the adjacent side *d*B of the angle *c*B*d*. It therefore follows, the centres of the circles being fixed, and their circumferences being moved simultaneously by contact at *c*, that the point *b*, in moving through half a revolution of its circle, will describe the right line *d*B, and therefore will describe the part *db* of that line in moving from *c* to *b*. If these points be joined, the angle at *b* will be a right angle, being in a semicircle; and, therefore, B*d* will be a tangent to the curve at that point, and *cb* a normal to it.

Let these conditions be considered organically. If an edge, coinciding with the curve *ab*, be fixed on the circumference of the contingent circle having its centre at A, and if B*d* be a radial edge movable on B, it is clear from the relations explained, that in moving from the point *c*, in which *a* and *d* primarily coincide, to the position depicted in the figure, the edge B*d* will be moved with the same uniform velocity by the curved edge, as if it were moved by a pin fixed in the point *b*, by which the curve *ab*, is supposed to have been defined. A normal to *b*, the point of contact of the edges, will moreover pass through *c*, the point of contingence of the circles, and therefore the angular velocity ratio will be the same as if the circles moved by the friction of contact. Were the curve *ab* of such a form that the normal did not cut the line of centres in *c* during some portion of its action, this condition would not be fulfilled; the angular velocity would increase in the axis within whose circle the normal fell; and it is easy to perceive, supposing the circles to be toothed, and to have the teeth so formed that in some portions of contact of the faces, the normal should meet the line of centres within the pitch circle of one wheel, there would be other portions of contact in which it would fall within the pitch circle of the other wheel, and therefore the motion would be irregular. For as the line of centres is divided into two parts at *c*, directly proportional to the number of teeth on the circumferences of the wheel and pinion, and as these circumferences have the same ratio as their radii, it follows that the arc of the pitch circle corresponding to each tooth of the wheel will be equal to the corresponding arc of the pitch circle of each tooth of the pinion; and as each tooth of the wheel must necessarily move, or be moved by, a tooth of the pinion, the arcs passed over by any two corresponding teeth of the pair must evidently be equal. But the circumferences moving through equal arcs in equal times, it is manifest that if there be instants during which the circle of the wheel moves quicker than that of the pinion, there must be other instants during which it moves slower; and as the point in which the normal meets the line of centres at any instant determines the angular velocity, it therefore follows that if there be some instants during which that point falls within the plane of the wheel, there will be other instants during which it will fall within that of the pinion, passing from one limit to the other through the point *c*, in which the velocity ratio will be a mean.

It thence appears, that the solution of the problem essentially depends upon the principle, that the teeth of one wheel must be of such a form that their outline may be described by the revolution of a curve upon the pitch circle of that wheel, while the outline of the teeth of the other wheel is described by the same curve revolving within its pitch circle. In the illustration given, this last is a straight line *db* in the direction of the radius; it is the part of the tooth within the pitch line, while the curve *ab* is the part without the pitch line of the other wheel. These parts will manifestly vary in length according to the length of the pitch arc; and they may be varied relatively, as we shall immediately see, but not individually; for in that case the point of contingence of the curves would be changed, and the normal would no longer be constant in the point *c*.

To complete the outlines of the teeth, according to the usual practice, the operation of finding the curves must be repeated on the interior of the pitch circle of the wheel, and on the exterior of that of the pinion, with a circle whose diameter is A*c*. The new curves thus generated, will obviously between themselves fulfil the same conditions as those depicted, when the pinion is made the driver. The normal to the point of contingence of the curves will, as before, pass through *c*; and therefore the new curves being added in continuation of the pair first formed, the origin of the normal will remain constant in *c* throughout the whole range of their possible action. It might also be shown, that being a normal to one point of contingence, and to one of the curves at that point, *cb* is necessarily a normal to every other point, and to both of the curves. To perceive that it is constantly a normal to one of the curves, let it be considered that when a circle rolls upon the circumference of another at rest, any point *b* in the circumference of the first, is at every instant of the revolution describing, during that instant, an arc of a circle about the point of contact *c* of the two circumferences; and therefore the right line joining these points is constantly the radius of the infinitesimal arcs described. But the whole curve is composed of the arcs thus described, and therefore *cb* is continually perpendicular to it; and we have seen that the conditions are not altered when the axes of the circles are fixed in position, and the motion of translation of the rolling circle compensated for by the motion of the base, thereby rendering the point of contact *c* also constant in position.

Reverting to fig. 31, let *p* be a point exceedingly near to *t* in the curve *nn*, which is fixed upon the circle B; it is evident from what has been shown, that as that point passes through the point of contact *t* of the two curves, it will be made to describe on the plane of the circle A an exceedingly small portion of the curve *mm*. But the curve which under these circumstances it describes, has been shown to be always perpendicular to the line *tc*; now the curve *mm* being perpendicular to that line at *t*, the point of contact with *nn*, that curve must likewise be perpendicular to it at the same instant, and consequently we have *tc*, a *normal* to both curves at the point *t*. This is the characteristic property of the two curves *mm* and *nn*, by which they satisfy the condition of a continual contact with each other at the same time that the circles revolve by the contact of their circumferences at *c*; and conversely supposing the motion to be induced by the mutual contact of the curves, they will communicate the same motion to the circles, as these would receive by the mutual contact of their circumferences.

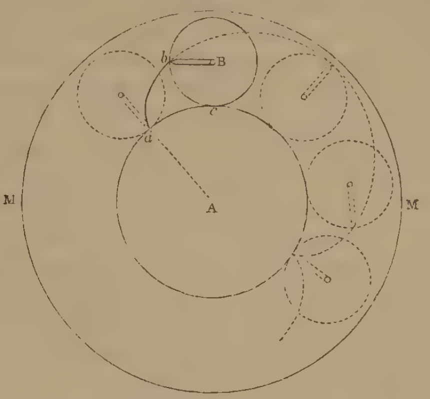
The principle here announced, exhibits a special application of one particular property of that curve known to mathematicians as the *epicycloid*. The mode of generating it is that described, and that upon which a mathematical definition of it is founded. Thus, generally, when two circles are in contact at their circumferences, and the one is made to roll upon the other, any point in the circumference of the moving circle describes during its revolution the particular curve named. This curve, as already stated, may be traced upon an immovable plane, against which a point in the moving circle is made to bear. For the sake of distinctness this is termed the *generating* circle, and the circle upon which it rolls is called the *fundamental* circle, and the portion of it on which the epicycloid rests is called the *base*.

The epicycloid, then, is a curve generated by a point in one circle rolling upon the circumference of another; it differs from the *cycloid* in this, that the latter curve is similarly generated by the rolling of a circle along a straight line. The cycloid may however be brought under the same definition by regarding the straight line as the circumference of a circle whose diameter is infinitely great.

The definition of the epicycloid is rendered obvious by reference to fig. 33, in which B is the generating circle, and A the fundamental circle of the epicycloidal curve *ab*. If the generating circle, instead of rolling on the outside, roll *within* the base circle, that is, on the concave circumference, the curve is usually called an *interior epicycloid*; that generated when the circle rolls on the convex circumference, being termed for distinction an *exterior epicycloid*. More commonly the last is denominated *the epicycloid*, while the interior epicycloid is sometimes termed *the hypocycloid*—the former being formed *above*, and the latter *below* the line of circumference of the fixed circle.*

An essential condition of the epicycloid is that the generating circle, in revolving from its first position to different other situations, as shown in fig. 33, applies successively all the parts of its circumference to those of the base; it is therefore evident that the base of the complete epicycloid is equal to the circumference of the generating circle, and each portion of its base, as *ca*, is equal to the corresponding part *cb* of the generating circle by the rolling of which it is traced.

Fig. 33.



From this we derive the modes of describing the epicycloid, which are represented by fig. 1, of Plate B, annexed. A series of equal arcs are marked off upon the base; and through these points, 1′, 2′, 3′, - - - a corresponding series of right lines are drawn from the centre O, extending beyond the circumference where the epicycloid is to be described, till they meet the arc EF, which defines the path of the centre C of the generating circle when the circumference rolls upon the base, but are not necessarily produced beyond the circumference when the hypocycloid is to be drawn. The upper and under parts of the figure are intended to exemplify the modes of proceeding adapted to these curves respectively. From the points *m*, *n*, *o*, - - - as centres, with radius = AC, let circles be described to represent the position of the generating circle as it rolls on the base from left to right. This done—from the points 1′, 2′, 3′, - - - set off arcs on the generating circle in these positions equal to the corresponding and equal arcs of the base measured from the same points to the origin A; that is to say, on the circle whose centre is *m*, set off the arc $1' \frown t$=the arc $1' \frown A$; on the circle whose centre is *n*, set off the arc $2' \frown u =$ the arc $2' \frown A$; on the circle whose centre is *o*, set off the arc $3' \frown v =$ the arc $3' \frown A$, and so on with the others, and the points, *t*, *u*, *v*, - - - will be points through which if a line be traced, it will be the epicycloid required, and will be more or less correctly defined according as the points thus found are more or less numerous, and therefore less or more distant from one another.

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* The epicycloid was invented by the celebrated Danish astronomer Römer, the discoverer of the progressive motion of light. He proposed this curve about the year 1674, as the proper form of the teeth of wheels in order to avoid rubbing friction. Newton gave the rectification of the curve in the first edition of his *Principia*, and Halley showed (*Philos. Trans.* No. 218), that its quadrature depends on that of the generating circle. The other properties of the curve were discovered and demonstrated by John Bernouilli—among others the remarkable property, that when the diameter of the generating circle is any aliquot part of the fundamental circle, the epicycloid is an algebraic curve, and is capable of being expressed geometrically in terms of the diameter of those circles. Thus taking the radius of the generating circle as unity, and that of the fundamental circle $= n$; then the length of the complete arc of the exterior epicycloid is expressed by the formula $8\left(\frac{n+1}{n}\right)$ and of the interior epicycloid by $8\left(\frac{n-1}{n}\right)$. In this case also, the epicycloidal areas are expressible in terms of the areas of the circles; the area of the exterior epicycloid being $= 3 + \frac{2}{n}$ times, and that of the interior epicycloid $= 3 - \frac{2}{n}$ times the area of the generating circle.

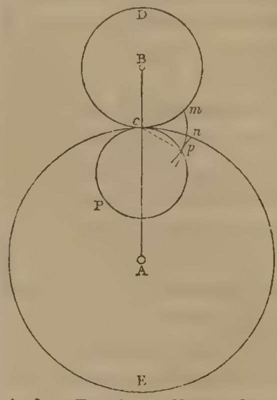
There is another mode of conducting the operation which is equally convenient, and is illustrated by the same figure. Let the series of arcs 1′, 2′, 3′, - - - of the base be also set off upon the rolling circumferences from the same point A where the circles meet. These divisions are marked 1, 2, 3, - - - in the figure. With one foot of the compasses in centre O, extend the other to 1, on the circumference of the rolling circle, and describe an arc through that point, meeting the radial lines OC and O*m* in *t* and *c*; again, with the distance of the point 2 from centre O, describe an arc meeting the lines OC and O*n* in *b* and *d*. Proceed in the same way to find the arcs *ae*, *rf*, *sg*, - - -; then from the point *b* set off *b* to *u* = the distance 2 to *d*; from *a* set off *a* to *v* = the distance 3 to *e*; from *r* set off *r* to *w* = the distance 4 to *f*; and so on, and the points *t*, *u*, *v*, *w*, - - - - will be as before points in the curve.

Both of these modes of construction are equally applicable whether the curve required be the external or internal epicycloid. The figure is intended to illustrate the process for both; but it may be observed in reference to the operation of laying off the arcs, that these ought to be taken by so small steps that the elementary portions shall not differ sensibly from their chords—which is the only measure that can be applied by the compasses. This is the more necessary, the greater difference there is between the diameters of the base circle and the generant; for the difference between an arc of given length and its chord being inversely as the radius of curvature, it is evident that the greater the length of two equal chords as of 1 to 2, and 1′ to 2′ of the generant and base, the greater will be the difference of the length of the arcs $1 \frown 2$ and $1' \frown 2'$, which they subtend in their respective circles. But the principle requires that the *arcs* be of equal length, and that the points 1 and 1′, and so on through as many of the divisions marked off as may be required, shall exactly coincide as points of contingence of the two circumference as they roll together from the point A.

We shall have occasion to point out a more ready means of finding these curves mechanically; but these illustrations of the geometrical conditions involved in their description, will assist in drawing attention to those properties upon which their applicability to our problem mainly depend.

To determine the fundamental conditions by which the two curves are mutually related in the teeth of wheels, let *mp* and *pn* be respectively portions of these curves, having the same generating circle *c*P*p*, and having for their bases the pitch circles *c*D*c* and *c*E*c* of two wheels. If the faces of the teeth upon the circumferences of these wheels coincide with these curves, it may be shown that they will work truly together; for let them be in contact at *p*, and let their common generating circle be in contact with both pitch circles at *c*, then will its circumference manifestly pass through the point of contact of the two teeth: and if it were made to roll through an exceedingly small angle upon the point *c*, rolling there upon the circumferences of both circles, its generating point would traverse exceedingly small portions of both curves. But since a given point in the circumference of the generating circle is thus at the same instant in the perimeters of both the curves, that point must of necessity be the point *p* of contingence; and since moreover the generating circle rolls upon the point of contact *c*, its generating point traversing a small portion of the perimeter of each of the curves at *p*, it follows that the line *cp* is a normal to both curves at that point; for whilst the generating circle *c*P*p* is rolling through an exceedingly small angle upon *c*, the point *p* is describing a circle whose radius is *cp*. If the teeth of a pair of wheels have their edges formed to these curves, they will therefore satisfy the condition that the line *cp*, drawn from the point of contact of the two pitch circles to any point of contact of the teeth, is a normal to the surfaces of both at that point, and this condition has been shown to be necessary and sufficient to the correct working of the teeth.

Fig. 34.



From this then it appears that if an epicycloid be described on the plane of one of the wheels with any generating circle, and with the pitch circle of that wheel for its base; and if a *hypocycloid* be described on the plane of the other wheel with the pitch circle of *that* wheel as *its* base, and if the acting surfaces of the teeth on the two wheels be cut so as to coincide with these curves, they will be driven by the intervention of these teeth in the same manner as they would by simple contact of their pitch circles.

It might be shown in exactly the same manner that the curves *mp* and *np* may be generated by the rolling of any other curve than a circle upon the pitch circles of the wheels; they would still possess this property, that a line drawn from any point of their contact to the point of contact of the pitch circles is a normal to both, which, as already shown, is the one necessary and sufficient condition. Further, it can be shown that, a tooth of any form whatever being cut upon a wheel, it is possible to find a curve which, rolling upon the pitch circle of that wheel, shall, by a certain generating point, traverse the edge of the given tooth; and the curve thus found being made to roll on the circumference of a second wheel, will trace out the form of tooth which will work truly with the first.

Let *a*M*b* of fig. 35, be any curve whatever intended to form the condition of the acting surfaces of the teeth of the wheel, and let *c*M be the pitch of the wheel; take *cm* = *c*M, it is required to find the curve which passes through the point *m*, and which being continually in contact with the curve *a*M*b* will be impelled by the latter according to the required condition. Now, as the common normal must always pass through the point *c*, if we draw the perpendicular *ct* to the curve *a*M*b*, which may be done by describing an arc of a

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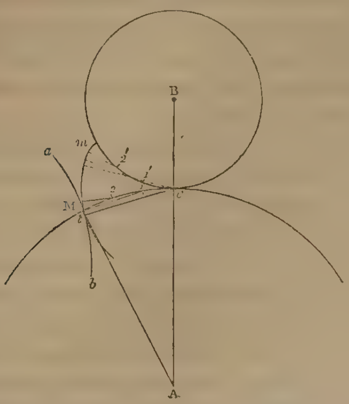
circle from the point *c* as a centre, with a radius such that that arc will cut the curve *a*M*b* in two points infinitely near and joining the point *c* to the middle of the distance of these two points, the point *t* will be the point of contact of the two curves. Supposing now that we divide the pitch *c*M and *cm* into the same number of equal parts, at the points 1, 2, and 1′, 2′, when these points arrive successively in contact, in the course of the revolution of the two circles, they will coincide with the point *c;* and the normal to the curve *a*M*b*, passing through the point 1, will also be the normal to the curve sought, and will pass through its point of contact with *a*M*b*. If then, from the point 1′ as a centre, with a radius equal to a normal to the curve *a*M*b*, we describe an arc of a circle, it will be a tangent to the curve sought. And doing the same with the other points of division, we find a number of points through which, if a curve be traced tangential to the small arcs described, it will fulfil the conditions of the problem, with more or less accuracy, according as the points taken have been more or less numerous.

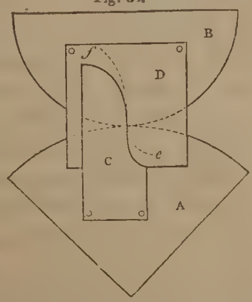
Fig. 35.



From this it therefore appears that, having the curve given of the teeth of one wheel, the curve to which the teeth of the second wheel ought to be formed may be readily found; which is the problem in its most general form, and may be stated summarily thus:—Given the form of the teeth of one wheel, to find out the form of those of another that may work with it correctly. *Solution:*—Describe the pitch circles of the required wheels, and find the curve which, revolving upon the one, will describe the given tooth; make the same curve revolve within the other, and with the same describing point it will generate the tooth required.

It is however to be observed, that these curves, to be applicable in practice, imply the condition that the curvature of the concavity of one tooth should be greater than the convexity of the other, or else that both should be convex. The practical solution may be conducted in the following simple manner:—Let A and B be two boards, whose edges are formed into arcs of the given pitch circles. Attach to one of them the shape of the proposed tooth C, and to the other a piece of strong paper, D, the tooth being slightly raised to allow the paper to pass under it. Keep the circular edges of the board in contact and make them roll together. Draw upon D a number of successive positions of the outline of the edge of C, which, touching one another in a corresponding number of points, a curve *fe*, passing through these points will be the corresponding tooth required for B. For it is obvious, from the method it is obtained, that if the tooth be cut out, and be made to work upon C, it will touch that tooth in every position; and therefore the contact of these curves will exactly fulfil the required condition, by replacing the rolling action of the two pitch circles.

Fig. 36.



This operation is more fully illustrated by fig. 2 of Plate B. In this figure let O be the centre of the piece B, supposed to be fixed; and suppose the piece A rolling upon it over the arc *cc*′, of sufficient extent to include the extreme positions of the acting surfaces; then, *c* and *c*′ being the extreme points of contact of the pieces, draw the radii O*c*, O*c*′, and from the centre O, with a convenient radius, draw an arc *de*, meeting the radii produced at *d* and *e;* divide this arc into a number of equal parts, corresponding to the number of positions of the tooth C intended to be taken; the radii drawn to the points of division will give corresponding points of contact at the circumference of the piece B, from which the relative positions of the tooth C may be severally described. In the figure, the extreme positions of the piece A and the tooth C are indicated by dot lines, though, with the exception of two other positions of the tooth, the rest is omitted to prevent confusion. A curve, *fe*, being traced so as to touch, in succession, the different positions of the tooth C, will indicate the form of tooth required for the piece B, so as to geer with the piece A.

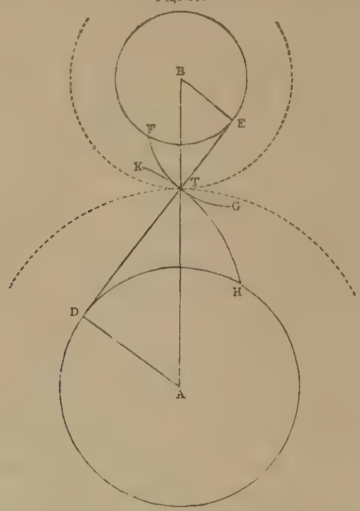
It is, however, true, that many forms of C may be tried which will prove impracticable; for some of the successive portions of its edge may come up and interfere with parts of *fe*, previously drawn; thereby showing, that although it may be geometrically possible to assign a form of *fe*, which shall work with any given form of C, it by no means follows that this is practically true; and, indeed, it does not appear that any new forms of curves deduced from this general principle are likely to adapt themselves in practice so well as those commonly adopted.

But there is a curve highly applicable in practice which we have not as yet noticed, and which at first may seem excluded from this general proposition. Its importance requires that it be noticed as a separate principle, which may be thus stated:—

The teeth of two wheels will also work truly together, if their acting surfaces coincide with curves traced out by the extremity of a flexible line, unwinding from the circumference of a circle, and called the *involute* of that circle, provided the circles, of which the bounding curves of the teeth are respectively portions, be concentric with the pitch circles of the wheels, and have their radii in the same ratio as the pitch circles.

Thus, let there be two centres of rotation, A and B, of which AT and BT are the primitive radii; through the point T draw any straight line, DE; through the points A and B draw the perpendiculars AD and BE, and describe circles from the points A and B as centres, with these lines as radii. Again, let KTH be the involute of a portion of the circle AD, traced by the (flexible) line DT, and this line is manifestly a normal to that curve at T; this point will also be that at which the second curve, traced by the line ET, touches the first. But suppose the circle AD to turn on its centre, and taking with it the curve KTH, the line DE will be a constant normal to that curve, in all its positions; and consequently also to the conducting curve in all *its* positions, and at the same point. Now this property of a curve, of having its normal tangential to a circle, belongs only to the involute of that circle; the conducted curve must therefore be the involute of the circle BE.

Fig. 37.



As an illustration of this principle, let DE be conceived to represent a band passing round the two circles drawn with the radii BE and AD; the wheels will evidently be driven by this band precisely as they would by the contact of their pitch circles, since the radii of the involute circles have the same ratio as the radii of the pitch circles. Let it also be supposed, that the projected circles carry round with them their planes as they revolve, and that a tracing point is fixed at any point T of the band; it will trace at the same instant the two curves, and upon both planes, as they revolve beneath it; it is then obvious that these curves, being traced by the same point, must be in contact in all positions of the circles when driven by the band, and therefore when driven by their mutual contact. The wheels would therefore be driven by the contact of teeth of the forms of the curves thus traced by the point T of the band, precisely as they would by the contact of their pitch circles; and it is easy to see that the curves HTK and FTG, so described, are involutes of the circles drawn with the radii AD and BE, from the given centres.[*]

A particular property of teeth formed according to the principle here pointed out, is that they transmit the pressure without altering its intensity—this being the same when we suppose the force constantly applied at the pitch circles, but not at the points of contact of the teeth. In epicycloidal teeth the normal to the curves is variable, as may be seen from fig. 31. Thus the pressure at the pitch circles being constant, may be expressed by A*c*, while that upon the points of contact of the teeth will be expressed by A*t*, which manifestly varies with the direction of the normal *ct*, and consequently with the position of the points of contact of the teeth. Such variation does not occur when the teeth are of the involute form; for then the relation of AD to BE is invariable, since the constant line DE is always normal to the curves in contact; and hence it is obvious that the pressure transmitted from the one circle to the other at D and E has always the same value.[†]

An advantage of this form of teeth is, that if the distance of the centres A and B be altered, but so that the involutes still remain in contact, the velocity of the pitch circles of the wheels will not be affected, and therefore the angular motion of the curves in contact will remain unaltered. This is a property which distinguishes the involute from the other curves that have been given, and is of some practical importance; for, in a pair of wheels of which the teeth are of that form, it is not only unnecessary to fix the centres at a precise distance, but a derangement of the centres, from wearing of the journals and brasses, or settlement of the framework, does not impair the action of the teeth; whereas in every other pair of curves which have been assigned, a variation of the distance of the centres destroys the equal ratio of the motion, by destroying the principle of their connection.

It may also be observed, that for every pair of pitch circles an infinite number of pairs of involutes may be assigned that will answer the required conditions; for the inclination of ETD to the line of centres is manifestly arbitrary, and every change of inclination manifestly produces a new pair of base-circles, and of involutes to these circles.

It has been intimated that the involutes may seem to be excluded from the general proposition whereby, having the curve of the teeth of one wheel given, a curve may be found for the teeth of another wheel which shall work correctly with it. This is not the case, although to prove the contrary would require an amount of analytic reasoning foreign to our present purpose. It may however be stated shortly in reference to the involute HTK, that the normal DT being inclined at a constant angle to the line of centres, and therefore to the tangent to the pitch circles at T, it may be generated under certain conditions by the revolution of a logarithmic spiral upon the pitch circle of that wheel; and similarly the revolution of the same curve within the second pitch circle will generate the other involute FTG, which, as already shown, will work correctly with the first.

SECTION VI.—APPLICATIONS OF THE FOREGOING PRINCIPLES TO THE FORMATION OF THE TEETH OF SPUR WHEELS.

1. *Of Epicycloidal Teeth.* The simplest illustration of the action of epicycloidal teeth, is when they are employed to drive a trundle, as represented in fig. 3 of plate B. In the first place, let it be assumed, that the staves of the trundle have no sensible thickness; that the distance of their centres apart, that is, their *pitch*, and also their distance from the centre of the trundle, that is, their pitch circle, are known. The pitch circles of the trundle and wheel being then drawn from their respective centres B and A, set off the

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[*] This concise illustration is due to Professor Moseley. See Mechanical Principles of Engineering, p. 254.

[†] The involute was first proposed by Euler, for the purpose of giving the form of the teeth of wheels, in his second paper on that subject. N. C. Petr. XI. 209.

pitches upon these circumferences, corresponding to the number of teeth in the wheel and number of staves in the trundle; let fine pins, *a*, *b*, *c*, &c. be fixed into the pitch circle of the trundle to represent the staves, and let a series of epicycloidal arcs be traced upon the pitch circle of the wheel, with a circle equal to the pitch circle of the trundle, and meeting in the points, *k*, *l*, *m*, *n*, &c. alternately from right and left. If now motion be given to the wheel in the direction of the arrow, then the curved face *mr* will press against the pin *b*, and move it in the same direction; but as the motion continues, the pin will slide upwards until it reaches *m*, when the tooth and pin will quit contact. Before this happens, the next pin *a* will have come into contact with the face *al* of the next tooth, which repeating the same action will bring the succeeding pair into contact; and so on continually.

We have here assumed the pins to represent the staves of the trundle and to be without sensible thickness, which is not true in practice. To allow, therefore, of the required thickness of staves, it is manifestly necessary and sufficient to diminish the size of the teeth of the wheel, by a quantity equal to the radius of the staves, sometimes increased by a certain fraction of the pitch for clearance, in this case termed *backlash*. This may readily be done by drawing within the primary epicycloids at the required distance, another series of curves parallel to these. For example, let it be required to draw the proper tooth to impel a stave of the radius *bt;* with that radius *plus* the requisite amount of clearance in the compasses, draw a series of small arcs *from* and *within* the original curve, then a curve touching all the arcs so drawn will be the epicycloid required; this curve acting upon the stave will fulfil the condition stated, for, being symmetrical with the first, which passes through the axis of the stave, it will act similarly at its circumference.

In practice, a portion must be cut from the points of the teeth, and also a space must be cut out within the pitch circle, of the driver to allow the staves to pass; but as the sides of these parts never come into contact with the staves, no particular form is requisite; the condition to be attended to is simply to allow of sufficient space for the staves to pass without contact.

We have here supposed the wheel to be the driver, and this being the case it is evident that the staves being indefinitely small, the contact of the tooth will begin at the instant its base reaches the line of centres at *a*; and during the action of the tooth the point of contact will gradually slide upwards, remaining always in the pitch circle of the trundle, at the same time it recedes from the line of centres until the contact is finally terminated at the point of the tooth. If the trundle be made the driver, the pitch lines of the pair will still move with the same uniform angular velocity; but the reverse of the preceding action will take place, for in that case the contact would commence at the top of the teeth and cease at their base; it would moreover commence before the line of centres, and terminate when it had reached the point of intersection of that line. Now the sliding which takes place between teeth whose point of contact *is approaching* the line of centres is of a much more injurious character than that which happens while the points of contact are receding from it. Not only does it cause much more friction and vibration, in consequence of the inequalities of the surfaces in contact—and the surfaces even of the most highly polished bodies have some inequalities, which, when pressed together, interlace—but the teeth at the same time tend to force the axes of the wheels outwards, thereby producing increased wear of the journals and injurious abraison of their own acting surfaces. When the action is receding from the line of centres, the friction is less intense, and its effects less injurious; it tends to draw the axes together, and induces much less vibratory action of the geering.

For these reasons it is studied in practice to avoid as much as possible the kind of contact which takes place before the line of centres, by making the wheel the driver and the trundle the follower. The diameter of the staves is also commonly made equal to that of the teeth with an allowance of a tenth of the pitch for clearance; the radius is therefore rather less than a quarter of the pitch; consequently the contact will begin, the wheel being the driver, when the centre of the stave reaches the line of centres, and therefore at a distance before that line equal to the radius of the stave, or rather less than a quarter of the pitch.

It is also evident that since one tooth must not *quit contact* before the succeeding tooth is engaged, the tooth T must still be engaged with the stave *b*, when the point *a* has reached the line of centres, and the point at which contact ceases must therefore be at an angular distance from the line of centres equal at least to half the distance *ar*, that is, to half the pitch. In a pair of this kind, the action which takes place before the line of centres is less than a half of that which takes place after passing it.

The action of a wheel and trundle being understood, it is easy to comprehend that of the teeth of a pair of wheels of the ordinary construction. Let A and B in fig. 4 of plate B be respectively the centres of a wheel and pinion of which the teeth are intended to be of the epicycloidal form, and A*c* and B*c* their primitive radii. To lay off the teeth of this pair, having determined the pitch and number of teeth in the wheel and pinion, let the pitch lines be divided into as many equal parts, setting out from the point of contact *c*, as there are teeth in them respectively. Let the thickness of the teeth be next set off, taking *ca* for the thickness of a tooth of the wheel, and *cb* for that of a tooth of the pinion. Upon the radii A*c* and B*c* as diameters, describe two circles, having also their point of contact at *c* and their centres X and Y. Now let the circle Y be made to roll upon the pitch line of the wheel, and a point in its circumference at *c* will describe the epicycloidal arc *cm*, and this curve determines the form of the point of the tooth of the wheel. In the same way describe the epicycloidal arc *cn* by making the circle X to roll upon the pitch circle of the pinion, and this curve will determine the form of the part of the tooth of the pinion beyond the pitch line.

Again, the teeth must have the same form on both sides; in other words, they must be symmetrical in reference to the radius which passes through the middle of their thickness, in order that the wheel and pinion may be turned in either direction. If then, through a point equidistant from *c* and *a* and from the centre A of the wheel, a right line be drawn as the axis of the tooth, it limits at *m* the outline of the tooth; to complete the form of the addendum, it therefore only remains to describe the opposite epicycloidal arc *am* equal to *cm*. A similar process completes the form of the addendum of the pinion.

The curve *cm* of the tooth of the wheel is constantly in contact with the radius B*c*, and its point of contact is at the same time situated in the circumference of the circle X; the contact will therefore cease when the extremity *m* of the tooth becomes the point of contact; and this occurs when the point *m* has arrived at the circumference of Y. If then an arc of a circle be described from the centre A with the radius A*m*, its point *f* of intersection with the circumference of Y is that at which the tooth ought to cease to act, to secure uniformity of motion; and at the same instant that another new tooth advances into geer. The determination of the point *f* limits also the useful length of the flank; for if from B as a centre, with radius B*f* an arc be described, the part *cs* of the radius *c*B is that which is in contact with the tooth till it arrives at the position B*f;* and consequently this is the useful length of the flank of the pinion.

Reasoning in the same manner with respect to the teeth of the wheel; it may be determined, that the useful length of the flank of the tooth *ca* is the portion *cg*.

We have yet to find the form of the portions of the teeth within their respective pitch circles, or more properly of the spaces between them. Now it has been shown that when the generating circle of a hypocycloid has its diameter equal to the radius of the base, the curve traced by it is a straight line passing through the centre of the base. If therefore the circle Y roll within the circle B, the tracing point in *c* will describe the right line *cs* (prolonged if required), in the radius B*c;* and this with the curve *cn* which meets it at *c*, and which, as above directed, is found by rolling the circle X upon the exterior of the same base, will complete one face of the tooth. And similarly the circle X being made to roll within the pitch circle of the wheel, will generate the straight line *cg* in the radius A*c*, thereby completing the face *mg* of the tooth of the wheel. The bottoms of the spaces may be defined by arcs described from the centres A and B, sufficiently within the pitch circumferences to allow the points of the opposite teeth to pass round without touching them: or they may be formed by inverse arcs described from the pitch lines.

In this instance, therefore, the flanks of the teeth of both wheels of the pair, that is, the portions of the teeth which lie within the respective pitch circles, are radial lines drawn from the centres to the pitch circles, and the faces of the teeth, or those portions which lie without the pitch circles, are arcs of epicycloids traced in each wheel with a describing circle equal in diameter to the pitch radius of the other wheel. Accordingly each flank and face will act in contact to produce a constant angular velocity ratio of the pitch circles of the pair, and the action of each pair of teeth will be confined to its own side of the line of centres.

In practice, and especially when the teeth of the wheels are small, it is not usually considered necessary to apply strictly the form of the epicycloid to find curves of the teeth; but to define them approximately by some such mode as the following.

Referring to fig. 5 of plate B, after having divided the pitch circles into the proper number of parts corresponding to the number of teeth, and set off upon the pitch lines the thicknesses of the teeth, through the extremity *b* of the pitch *ab* of the pinion, draw a radius B*b* cutting the circumference of the epicycloidal circle Y in *d*. Joining the point *d* with the first division *b′* of the wheel, draw a perpendicular upon the middle of the line *db′* meeting the circumference of A at the point *h;* an arc described from this point as a centre, and with *hb* as a radius, will define the curve of the teeth of the wheel with sufficient exactness.

The length of the teeth is limited at *d* where the curve of the tooth meets the circumference of the epicycloidal circle Y; for when the tooth has arrived at that point, another pair ought to come into geer at the line of centres. The flank of the tooth being radial, may be defined by a right line from the termination of the curve of the face of the tooth in the pitch circle drawn towards the centre A.

As the tooth is symmetrical about the axis A*k*, its opposite face may be described by taking off the form of the side found upon a mould; and the two sides being thus found, a pattern tooth may be formed and used in drawing in the other teeth of the wheel.

A similar process gives the form of the teeth of the pinion. Having taken an arc *ae* equal to the pitch, draw A*e* till it meets the circumference of X in *g*; with this join the division *e′* of the pinion, and through the middle of the line draw a perpendicular, meeting the circumference of Y in *h′*; from this point as a centre and with radius *h′e′* describe the curve *ge′*, which defines the face of the tooth. A radial line meeting the curve at *e′* and drawn towards B as a centre, will in like manner define the flank of the tooth.

To determine the bottom of the space between two teeth, it is to be remarked that in the pinion this must be at a distance from the centre A, at least equal to A*n;* if therefore from the point B as a centre, with radius B*m* a circle be described it will define the bottoms of the teeth of the pinion; and in like manner if with radius A*n* a circle be described from the centre of the wheel, it will define the bottoms of its teeth. The angles at bottom are commonly rounded—which, at the same time that it improves the appearance, strengthens the teeth.

The forms of the teeth are also occasionally described by arcs, of which the radii are equal to the pitch, and with the centres taken upon the pitch lines.

In this case, the centres of these arcs must fall upon the sides of the teeth, which being already marked off, it becomes a simple process to describe the upper portions. This method, which differs little from the preceding, may be conveniently substituted for it, when the diameters of the wheels are not very unequal, and the teeth not thick. But for small pinions with larger teeth, the first method must be adopted.

In many cases the curves of the teeth are described by arcs drawn from the centres of the adjacent teeth upon the pitch line. This gives a radius equal to the pitch, *plus* half the thickness of a tooth. In like manner, the sides of the teeth also are sometimes described by arcs from the centres of the adjacent teeth giving a radius equal to the pitch, *minus* half the thickness of a tooth.

This form of the tooth may be defective, in the case of a very small pinion, having to transmit great pressure, as the extremities of the teeth may be too much reduced. In this case, the curves or faces of the teeth may be described with radii equal to $\frac{3}{4}$ths of the pitch; and if this be not sufficient curvature, radii equal to some smaller fraction of the pitch may be used.

When, on the contrary, the pitch is large, and the pressure comparatively small, the teeth may be too short; this will be remedied by employing arcs of which the radius is one and a half or twice the pitch.—A general method of determining these radii will presently be given.

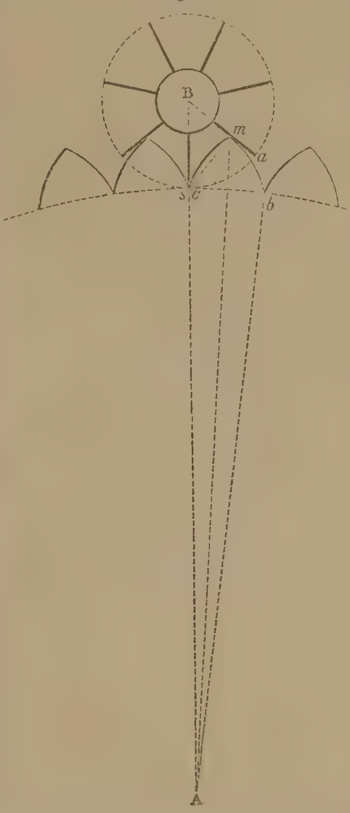
Another condition which it is necessary to advert to, is the point in the circumference at which contact of the teeth commences. This point is necessarily that in which the two pitch circles cut the line of centres, as may be rendered obvious by reference to fig. 4 of plate B. Here it is plain that the face *cm* of the tooth of the wheel ought to act upon the side of the tooth of the pinion, until the extremity *m* of the tooth arrives at the extremity of that side, as shown by the position of the tooth *f*. But were this to take place before the acting surface of the succeeding tooth of the pinion had arrived at the line of centres, the succeeding tooth of the wheel would of necessity act, during the first part of its contact, upon the face of that of the pinion, and continue so to do until it had arrived at the line of centres.

On the other hand should the extremity of the tooth, on arriving at *f*, not immediately cease contact, but only be disengaged when the acting surface of the next tooth of the pinion had arrived at or passed the line of centres, the *face* of that tooth would not be at all acted upon by the tooth of the wheel, and the contact with the flank would be entirely beyond the line of centres. In this case, were the wheel constantly to drive the pinion, the curved faces of the teeth of the latter might be altogether dispensed with, without in any way injuring the equality of action. But the condition assumed does not occur in practice; for on account of the inequalities which take place in the motion of the wheels—whether arising from inequality in the resistance, or of speed in the moving power, the wheel and pinion act upon each other alternately, even when the general movement is continuously in the same direction—a circumstance which cannot be safely neglected in the construction of any system of geering.

When the teeth fall into action *before* the line of centres, they must obviously slide upon each other as they approach that line, and their friction, as in the case of the wheel and trundle, is then much greater than it is after they have passed it, on account of their sliding *inwards* upon each other instead of *outwards*, and against a pressure which is augmented in proportion to the curvature of the acting surfaces. And moreover, in the contact *after* the line of centres, although there exist inequalities on the acting surfaces, they offer comparatively little obstruction to the sliding of the teeth upon each other; whereas, when the contact is in advance, every inequality becomes an abutting obstacle to the *closing-in* of the teeth, and which it is necessary to surmount by a species of backward motion at great expense of motive power, and greatly to the detriment of smoothness of motion among the parts of the geering.

But although it is thus clear that an advantage is gained by making the teeth engage after the line of centres, yet this is as clearly desirable only within certain limits. For as already observed, it is necessary that the tooth of the pinion continue to geer with the wheel until the succeeding tooth has arrived *at* the line of centres—a condition in which is involved the problem of the *minimum* number of teeth that can be given to a pinion to work with a wheel of given size, and which may be determined when the pitch and radii are given. From these data, the angle of the sides of the teeth is immediately deduced. The length and thickness of teeth and width of space must be modified to answer the condition imposed, and the smaller the number of teeth in the pair, the more difficulty there is in accomplishing the solution. To resolve the conditions of the question into the simplest form which they admit, let us take the imaginary case of a pinion of *seven* teeth without visible thickness, to geer with a wheel of *fifty* epicycloidal teeth as represented in the annexed diagram. In this the line AB is the line of centres; *cmb* one of the teeth of the wheel, and *Bma* one of the radial teeth of the pinion, with which the tooth *cmb* has been in contact during its motion from *c* to *b*. Now since the pinion has seven radial teeth, the value of the angle *c* B*a* is $\frac{360}{7}^\circ = 51^\circ\ 25'\ 43''$; and making the radius B*c* =1, the right-angled triangle B*mc*, completed by drawing the right line *cm* gives,

Fig. 38.

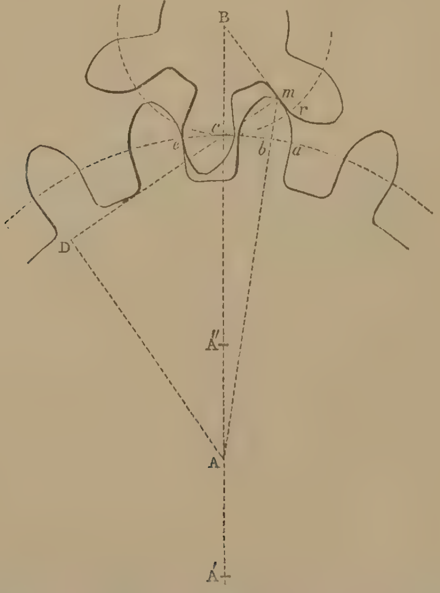


$$Bm = \cos aBc = \cos 51^\circ\ 25'\ 43'' = 0.62349.$$

If radius B*c* = 1, then A*c* = $\frac{50}{7}$ and AB = $\frac{57}{7}$. Thus we have given in the triangle A*m*B the two sides AB and B*m* and the angle *a*B*c*; and therefore we have also the angle BA*m* = 3° 35′ 50″. Now if the wheel have 50 teeth, we have the angle *c*A*b* containing a tooth and space, = $\frac{360}{50}^\circ$ = 7° 12′; and subtracting from this the angle *c*A*m*, there remains the angle *b*A*m* = 3° 36′ 10″; but the angles *b*A*m* and *m*A*c* are equal, and by consequence the angle *b*A*c* is double the former of those, that is, 7° 12′ 20″, the tooth being symmetrical in reference to A*m*. Now we found the angular value of the pitch *cb* = 7° 12′, which ought to contain the tooth and the space. It therefore follows that even when the tooth of the pinion has no thickness, it is impossible to place the tooth of the wheel in the pitch interval—for this is shown to be less than the tooth of the wheel would occupy; and it is easy to understand that a wheel of less than 50 teeth could still less be made to drive a pinion of 7 teeth. And although in a wheel which has more than 50 teeth, the pitch angle would be found greater than the thickness of a tooth, still it would be less than that which should contain the thickness and the space together; hence the angle which would be found due to the space would be so small as to be insufficient to receive the tooth of the pinion allowing for lateral clearance, and in consequence the pair would necessarily geer *before* the line of centres.

If we repeat the preceding calculation for a pinion of 8 teeth, driven by a wheel of 57 teeth, we shall find that the angle $sAc = 1^\circ\ 11'\ 17''$ while the angle of the thickness $cAb = 5^\circ\ 7'\ 40''$;—showing that the angle of the space is not sufficiently great to admit a tooth of reasonable thickness; and it is plain that the difficulty would be increased by giving a less number of teeth than 57 to the wheel. In like manner with a pinion of 9 teeth, and a wheel of 64, the angle of the space $= sAc = 1^\circ\ 51'\ 48''$ while the angle $cAb = 3^\circ\ 45'\ 42''$; and consequently the condition cannot be fulfilled of making the pair geer only at and after the line of centres. But if we repeat the calculation with a wheel of 72 teeth and a pinion of 10, we shall find that they very nearly satisfy the required condition, for with these numbers the angle of the space $sAc = 2^\circ\ 12'\ 44''$ while the angle of the thickness $= cAb = 2^\circ\ 47'\ 16''$.

Fig. 39.



This problem is often more conveniently solved organically, when the teeth of the wheel and pinion are formed in the usual manner. Thus in the annexed figure, let cBm be the angle through which it is desired that the contact of the tooth am should continue after passing the line of centres; then as the contact ends at m, the point of contact will be at the extremity m of the tooth. Join cm by a right line which will be a perpendicular to the radius Br; also join Am. Then since a was in contact with r at the line of centres, the arc $ca = cr$ is given, and is that portion of the pitch through which the contact of the tooth is required to continue. Then ab is half the thickness of tooth, *minus* half the base of the portion cut off; and ea is equal to the pitch and must contain one tooth and the space between. Supposing these quantities to be equal, then since ab contains less than half a tooth, and cannot contain more, eb must contain half a tooth and space at least, and in this case as much more as ab is less than half a tooth. Therefore for every given value of cBm, a value of cA may be assigned which shall make eb exactly equal to a space and half a tooth, when the tooth is pointed, and with a corresponding and known increment when the tooth is blunted.

If a greater radius cA' be taken, the point b will manifestly fall still nearer a and the tooth may be still more blunted, but if a less radius cA'' be assumed, then the point b will fall nearer to c, and eb will become too small to contain the space and remaining half tooth; and in this case, if the wheel were set out it would be found that the epicycloidal arcs on the two sides of the tooth would intersect between m and b, and therefore the tooth would be too short to continue in action through the required arc ca.

This mode of illustration is due to Professor Willis, to whom we are also indebted for the following tables derived organically in the manner described, for spur and annular wheels:—

Table I.—For spur wheels, showing the least numbers of teeth that will work with given pinions—Tooth = space.

	No. of teeth in given pinion.	Least Number of teeth in the wheel.	
		Wheel driving.	Pinion driving
Arc of action cr = pitch.	5	impossible.	impossible.
	6	—	176
	7	—	52
	8	—	35
	9	—	27
	10	rack.	23
	11	54	21
	12	30	19
	13	24	18
	14	20	17
	15	17	16
	16	15	—
$cr = \frac{3}{4}$ pitch.	3	impossible.	impossible.
	4	—	35
	5	—	19
	6	—	14
	7	31	12
	8	16	10
	9	12	10
	10	10	10
cr = 2-3d pitch.	2	impossible.	impossible.
	3	—	36
	4	—	15
	5	—	13
	6	20	10
	7	11	9
	8	8	8

Table II.—For annular wheels, showing the greatest numbers of teeth that will work with given pinions—Tooth = space.

	Number of teeth in pinion.	Greatest No. of teeth in annular wheel.	
		Wheel driving.	Pinion driving.
Arc of action cr = pitch.	2	impossible.	5
	3	—	12
	4	—	26
	5	—	85
	7	14	any number.
	8	25	—
	9	60	—
$cr = \frac{3}{4}$ pitch.	2	impossible.	10
	3	—	77
	4	5	any number.
	5	12	—
	6	77	—
cr = 2-3d pitch.	2	impossible.	14
	4	8	any number.
	5	64	—

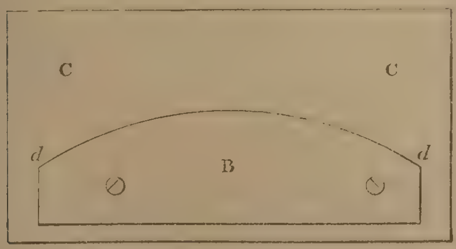
These tables contain all the limiting cases under the three suppositions of the arc of action being equal to the pitch, to *three-fourths* of the pitch and to *two-thirds* of the same. When the arc of action and pitch are the same, the teeth of the follower may be reduced in length to the pitch circle, and the contact of the teeth confined to their recess from the line of centres; when the arc of action is reduced, as in the other two cases given, the reduction will of course indicate the portion of contact which must take place in advance of the line of centres. Thus when $cr = \frac{2}{3}$ and $\frac{3}{4}$ of the pitch, then it is obvious that the contact of the teeth must begin at least $\frac{1}{4}$ and $\frac{1}{3}$ of the pitch respectively before they reach the line of centres.

It also appears from the table that a smaller pinion may be employed to drive, than to follow. Thus when the action begins at the line of centres, the least wheel that can drive a pinion of *eleven* teeth is *fifty-four*; but the same pinion can drive a wheel of *twenty-one* and upwards. Again, nothing less than a rack can drive a pinion of *ten*, but this pinion can drive a wheel of *twenty-three* and upwards. No pinion of less than ten leaves can be driven, but pinions as low as six may be employed to drive any number above those in the table. And lastly, the least pair of equal pinions that will work together is 16. These limits being geometrically exact, and making no allowance for wear, it is better, indeed necessary, in practice to allow more teeth than the table assigns.

It often occurs in practice that a wheel is required to drive more than one pinion, and *vice versâ*, and these of different diameters. In this case the usual mode of obtaining the epicycloidal form of the face of the driving teeth is inapplicable, as that curve is made to depend upon the diameters of the pair when uniformity of motion is studied. But before adverting to the methods by which this condition may be fulfilled, it will relieve us of some of the difficulties of the question to examine the ordinary mode in which epicycloidal teeth are set out. As the workman cannot be expected to be in possession of that accurate mathematical knowledge, which would enable him to determine the necessary forms by purely theoretical means, he has recourse to the mechanical method of forming the pattern teeth by *templets*. We give the process in detail.

The pitch of the teeth being given, and the ratio of the angular velocities, the diameters of the wheels are determined to the nearest numbers of teeth which correspond to these conditions. Four slips of thick veneer are then provided; on two of these, small arcs *dd* of the pitch circles are drawn, and on the other two, similar arcs of circles whose diameters are equal to the radii of the wheels. The edges of the pieces towards which these arcs are convex, are worked accurately to the arc-lines; and thus prepared, the first pair B, representing segmental portions of the wheels at their pitch circumferences, are fixed by small screws upon the faces of two pieces of clean hardwood board CC.—This completes the pair of templets, as represented by the annexed cut, fig. 40.

Fig. 40.



The operation of describing the form of the teeth is thus conducted: The thickness *ac* of the tooth being set off upon the pitch circle *dd*, the segment D, *whose diameter is equal to the radius of the opposite wheel*, is then applied convexly against the edge on *dd*. This segment has a tracing point inserted obliquely into its edge at *p*, which being brought to coincide with *a*, in the pitch line *dd*, and the segment made to roll towards the right, the edges meantime being lightly pressed together, the tracing point describes the curve *at* upon the surface CC. To describe the side *ct*, the point *p* is made to coincide with *c*, and the segment D is rolled towards the left, till the curves intersect at *t*. To prevent the segmental edges sliding upon each other during the operation they are sometimes rubbed with chalk or flour-rosin.

Fig. 41.

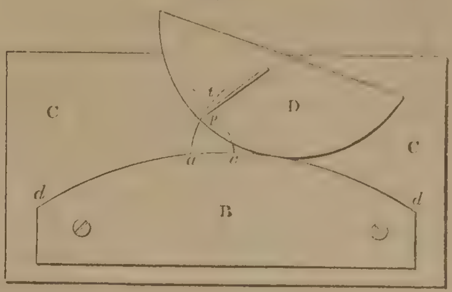
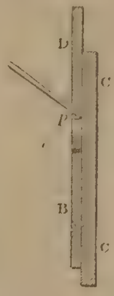
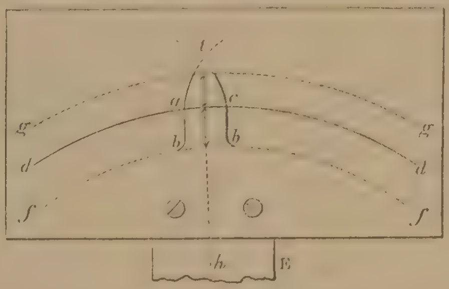


Fig. 42.



The curvilinear faces of the tooth being thus defined, the portion *dd* of the pitch circle is traced upon the plane CC, and the segment B is removed; the board CC is then fixed upon a slip of deal E, something more in length than the radius of the wheel; and the arc *ac* being bisected, the straight line *ht* is drawn through the point of bisection, and prolonged in the radius to the centre of the wheel on E. On the line *th*, beyond the pitch circle, the length of the addendum of the tooth $=\frac{3}{10}$ of the pitch, is set off, and through the point of limit, which will always fall within *t*, the arc *gg* is drawn from the centre of the wheel. Similarly the length of the tooth within the pitch circle is set off $=\frac{4}{10}$ of the pitch, and through that point of limit the arc *ff*, is struck to define the bottoms of the spaces between the teeth. The flanks *ab*, and *cb*, are radial lines to the centre of the wheel; and these being drawn, and their angles of junction with the bottoms of the spaces being slightly filleted, to increase the strength of the tooth, and render the pattern more easily moulded, the outline of the tooth is considered complete. The angles formed by the meeting of the arc *gg* with the sides of the curves *ta*, *tc*, are sometimes rounded off agreeably to the notion that the tooth will thereby in working have more facility to enter the spaces of the opposite wheel; but it is easy to perceive that this modification of the form is simply equivalent to shortening the face curves to that extent, for presuming the tooth to be a driver, it

Fig. 43.



will evidently quit contact so much sooner; in other words, its arc of action beyond the line of centres will be proportionably shortened.

The form of the teeth of the opposite wheel is found exactly in the same manner, the pitch segment of that wheel being employed as a base, instead of the segment B; and the segmental piece D, being replaced by the segment upon an arc of half the radius of the wheel, whose form of tooth has just been determined.*

These operations give correct working forms of the teeth of a geering pair; but the method is faulty, in as far as it gives a form of tooth smaller at the root than at the pitch circle; and also in the circumstance that neither of the wheels of a pair thus set off will work correctly with a wheel of another pair of the same pitch in which the ratio of the angular velocity is different. For example, a wheel of 30 teeth set off to work with another of 50 teeth will not work correctly with a wheel of another pair of which the numbers are 40 and 50; and the farther the ratio recedes from 30 : 50 the greater will be the irregularity of motion.

But in the case of spur geer it is intended that any wheels of the same pitch shall work with any others, however different in numbers. To fulfil this condition, it is necessary to employ the proposition explained at p. 100, from which it appears that if there be two pitch circles touching each other, an epicycloidal tooth formed by causing a given circle to roll on the exterior circumference of the one, will work correctly with a hypocycloidal tooth formed by causing the *same* circle to roll on the interior circumference of the other. This proposition is as well established and quite as obvious as that upon which the common mode of proceeding is founded; and where it has been admitted into the workshop as a fundamental truth, it is found to be attended with even greater facility of application than the ordinary method.

Fig. 45.

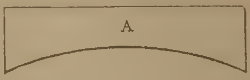
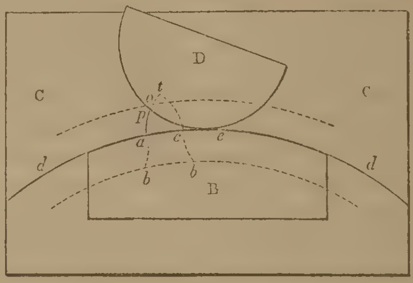


Fig. 46.



To describe the practical application of the principle, let a thin slip of wood be provided, and let an arc of the pitch circle be struck upon it; divide the slip into two portions through the line of this arc with a fine saw; one part, A, will have a *concave*, and the other, B, a corresponding *convex* circular edge. (Or the same slip may be made with a convex and concave edge of the same radius.) Describe an arc *dd* of the pitch circle upon a second board, CC, upon which the pattern tooth is to be drawn. Fix the piece B upon the board, so that its circular edge may accurately coincide with the circumference of the arc *dd*. A portion of a circular plate D is next provided, of the same radius which it is proposed to give to the generating circle: this plate has a fine tracing point at *p* inserted into it, and projecting slightly from its under surface, and accurately coinciding with its circumference. Having set off the thickness of the tooth *ac* upon the pitch circle *dd*, so that twice this width increased by the clearance which it is desired the teeth should have, may be equal to the pitch, the generating circle D is made to roll upon the convex edge of B; meantime the point at *p* will trace upon the board the curve of the faces of the tooth, having caused the point to coincide successively with the two points *a* and *c*, and the circle to roll from right to left, and *vice versâ*.

Let the piece B be now removed, and the piece A applied and fixed, so that its concave edge may accurately coincide with the circular arc *dd;* then, with the same circular plate D pressed against the concave edge of A, and made to roll upon it, the point at *p*, which is made as before to coincide successively with the points *a* and *c*, will trace upon the surface of the board CC the two hypocycloidal arcs *ab*, *cb*, which form the flanks of the tooth. The complete tooth, thus formed, has this advantage over the form of tooth found by the former mode, that it will work correctly with the teeth similarly described upon any other wheel, provided the pitch of the teeth be the same, and provided the same generating circle D be used to strike the curves upon the two wheels.

In this manner the general forms of the teeth of the pair

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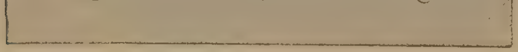
* The erroneous practice of describing the outlines of the teeth, by rolling the templet of one wheel upon that of the other, is even yet pertinaciously retained where it might be least expected. The mode of proceeding is represented in the annexed cut, of which C is the

Fig. 44.

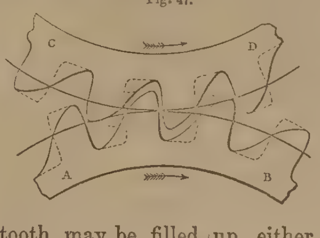


templet of the pinion, and A that of the wheel, the arcs *g* and *f* being respectively their pitch circles. The templets are formed in the same manner as above described, except that sometimes the planes upon which the pitch arcs are fixed, are cut to the circles limiting the points of the teeth, (as shown by the cut), but not necessarily. The drawing points *m*,*m*, are inserted obliquely, (as represented in fig. 42 above) into the edges of the pitch and diameter circles of the pinions; and in this condition the templets being made to roll together through a sufficient arc, the curves *r* and *v* are described by the points. The former *r*, of these curves is that of the face of the wheel-teeth, and falls upon the space B, equal in breadth to the addenda of the teeth; and the latter *v*, traced on the plane of A, by the point in the edge of the addendum circle of the pinion, is the outline of the flank. The operation being repeated, with the tracing points inserted into the templet of the wheel, gives the curves of the teeth of the pinion on *its* templet.

are determined. And it only remains to cut them off at such lengths that they shall come into contact in the act of passing through the line of centres. Thus, in fig. 4 of Plate B, let Y be the centre of the generating circle of the teeth, then the points in which that circle intersects the edges of the driving teeth, are the points of contact of the teeth. Now as the teeth ought to come into action only at the line of centres, it is clear that the tooth *f* must have been driven by the tooth of the wheel from the time their contact was at *c*, to which another tooth of the wheel is shown to have advanced, the former being about to quit contact; and since the tooth now advanced to *c* is about to take up the task of impelling the driven wheel, and the other to yield it, it is clear that all the part of the tooth falling within the circumference of the generating circle might be removed; this done, it is evident that all the driving strain would immediately be transferred to the succeeding pair of teeth as required. In order, therefore, to obtain a form of tooth which shall satisfy this condition, by the mode of setting out which has just been described, it is only necessary to take *ae*, in fig. 46, equal to the pitch of the tooth, and to bring the circumference of the generating circle D to touch the convex circumference of the arc *dd* in that point: the point of intersection *o* of this circle, with the face *ta*, will be the last acting part of the tooth; and if a circle passing through that point be struck from the centre of the pitch circle, all that portion of the faces of the teeth which lie beyond it may be cut off. The length of the tooth of the wheel intended to act with this may be determined in exactly the same manner. It may, however, happen that the point *o* will fall beyond the point *t* of the tooth; when this occurs, it is then impossible to cut the teeth of such length as to satisfy the condition that they shall drive only after the line of centres. This may be proved organically with a pair of 15 to 15, and other numbers which the preceding table of cases will suggest.

We have here supposed the same generating circle to be used in striking the entire surfaces of the teeth of both wheels. This is convenient, but not essentially necessary; for if the same generating circle be used in striking those parts of the teeth which act together, the required condition will be satisfied. Thus the *flank* of the driving tooth and the *face* of the driven tooth come into actual contact: it is therefore requisite that these curves be respectively an epicycloid and hypocycloid, struck with the same generating circle. And again, the *face* of the driving tooth and the *flank* of the driven tooth being in contact, these curves must in like manner be struck with the same generating circle. But it is evidently unnecessary that the same generating circle be used in the first and second cases; any generating circle will satisfy the proposed conditions in either case, provided it be the same for the epicycloid and the hypocycloid, which are to act together; observing, however, that as the diameter of the generating circle is increased, the thickness of the root of the teeth will be diminished; and conversely, when made *less* than the radius of the pitch circle, the root of the teeth will spread out, and the curvature of the epicycloidal faces will be proportionally increased, and the teeth shortened. From these observations it is also clear that the forms of the teeth of a pair of wheels which are required to drive continuously in one direction may be so modified as to give additional strength to the teeth without impairing their delicacy of action. Thus, if the wheels represented here, the lower AB being the driver, move always in the direction of the arrows, then, instead of the teeth being formed symmetrically, the back of each tooth may be filled up, either by striking the curves by a very small generating circle, or better by involutes, which being proportional to the pitch circles will be sure to clear during the working of the wheels. By this means the strength of the teeth is greatly augmented, as must be obvious, and their action is in no way impaired, since the acting surfaces are not altered.

Fig. 47.



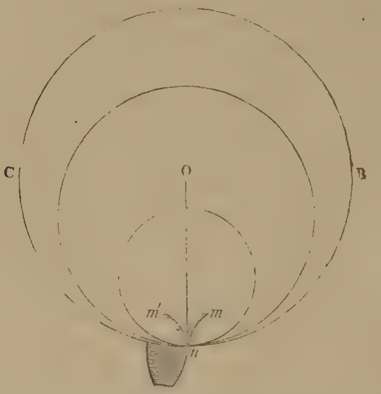
Returning now to the condition often required, though seldom fulfilled in practice, of making wheels of different numbers, when of the same pitch, to work correctly together: it is clear from what has been advanced in connection with the preceding mode of *setting out*, that it is not absolutely necessary to assume the generating circle of any precise radius; it may be taken greater or less within certain limits without affecting the correctness of the working surfaces of the teeth. This premised, we are in a position to understand the following principle, first announced by Sir David Brewster, and which forms the basis of an excellent paper on the teeth of wheels by Prof. Willis, originally published in the Transactions of the Institution of Civil Engineers, and subsequently in his Treatise on the Principles of Mechanism. The proposition is this:

Proposition.—"If for a set of wheels of the same pitch, a constant describing circle be taken, and employed to trace those portions of the teeth which project beyond each pitch line by rolling on the exterior circumference, and those which lie within by rolling on the interior circumference; then any two wheels of this set will work correctly together."

In illustration of this principle, the author adduces the established and well-known condition, that the portion of each tooth *within* the pitch line of a driving wheel works only with the portion which lies beyond the pitch line of the follower, as above shown, and that its action is confined to the approach of the point of contact to the line of centres. After the point of contact of the teeth has passed that line, then the case is reversed, and the portion of the driving tooth which lies *beyond* the pitch line is in contact only with some part of the follower's tooth, which lies *within* its pitch line. Consequently, if a constant describing circle be used for the whole set, it is clear that the proposition will apply to any pair of the wheels, both before and after the teeth have passed the line of centres; for in each case we have an exterior epicycloid working with an interior epicycloid, and both have been drawn by the same describing circle; that is, by the constant circle of the set.

To carry this scheme into practice, it only remains to settle the proper diameter to be given to the describing circle. This may be done by considering the effect this diameter has upon the form of the tooth, a subject already dwelt upon, but which may be further illustrated by the relation embodied in the annexed diagram, which is that employed by the author. Let the external circle BC be a pitch circle whose centre is O; then upon this system the flank of the tooth—that is, the portion of it which lies within the circle—will be an arc of an interior epicycloid (or hypocycloid) *mn* or *m'n*. Now, if the describing circle be half the diameter of the pitch circle, the flank will become a straight line, coinciding with the radius O*n*, as shown in figs. 4 and 5 of Plate B. But if the describing circle be of less than half the diameter of the pitch circle, the flank *mn* will be concave, and the base of the tooth will spread; on the contrary, if the describing circle be more than half the diameter, the flank *m'n* will be convex, and the base of the tooth will be contracted inwards—a form manifestly unpractical and useless. It is therefore clear, that the describing circle must not be greater than half the diameter of the pitch line; nor must it be too small, for then the base of the teeth would spread inconveniently, and the curvature of the exterior epicycloids would be injuriously increased, and the teeth become too short. The circle selected should therefore be as large as it can be, consistently with the limitation just stated, so that we finally arrive at the practical rule: *Make it equal in diameter to the radius of the least pitch circle of the set.* And, as pinions should never have less, if possible, than 14 teeth, it would be convenient to take the pitch circle of that number in all cases as the generant of the set.

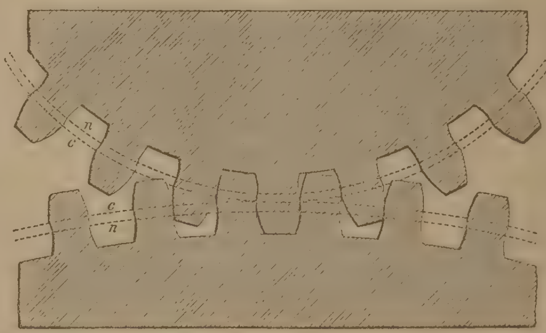
Fig. 48.



By the use therefore of the same generating circle for all the wheels of the same pitch, they will all work correctly together; so that when the same wheel is required to geer with two others of different numbers, the motion will have all the accuracy which can be commanded for a single pair set off to each other by the usual modes.

We have had occasion to point out some approximate modes of describing the tooth-curves initially by means of the compasses. These are generally, however, considered to offer so little advantage in point of facility over the ordinary mechanical procedure by means of the templets, that they are rarely resorted to in practice. But when the primary arcs have been found by the templets, the ordinary course—except when the teeth are very large, and then pattern shapes* are usually cut out and employed to trace the outlines of the forms by—is to find by trial the centre and length of radius to which a circular arc may be described, approximating as closely as possible to the given epicycloidal arc, and with these to *lay-in* the teeth of the wheel. The arcs are usually found by simply taking any three points in the given curve, and determining the radius with which a circle may be drawn, whose circumference will pass through the points so taken. Should the position of the centre of curvature of these arcs deviate from the pitch circle, a circle is struck on the ends of the unformed teeth, concentric with the first, and passing through the point in which the radius of curvature terminates. Thus, in the annexed figure, *c*, *c*, are the pitch circles, and *n*, *n*, circles struck concentric with them, though the centres of curvature of the flank curves—which in this case are hollow to the radii. It as often happens that the required circle falls without the pitch line; but the principle and mode of proceeding are the same. When the required circles are struck and the compasses adjusted to the arc radius to be employed, the pitch divisions and thickness of the teeth being of course laid off on the pitch line, the lateral tooth-lines are described on the blank cogs in succession. Thus in the figure given, the flanks, as already intimated, are drawn from centres through which the circles *n*, *n* pass, and the faces of the teeth from centres which coincide with the pitch circles *c*, *c*; but all these curves have their different radii, which are previously determined, and with which individually the arcs are drawn-in by stepping the compasses from cog to cog round the entire pattern of the wheel, laying in the arcs in succession until the outline of every tooth is complete.

Fig. 49.



The great practical convenience of this method is evinced by the extent of its adoption, and requires only to be aided by a more commodious and certain method of determining the centres and radii of the approximate arcs. Euler was the first to undertake the necessary investigation;

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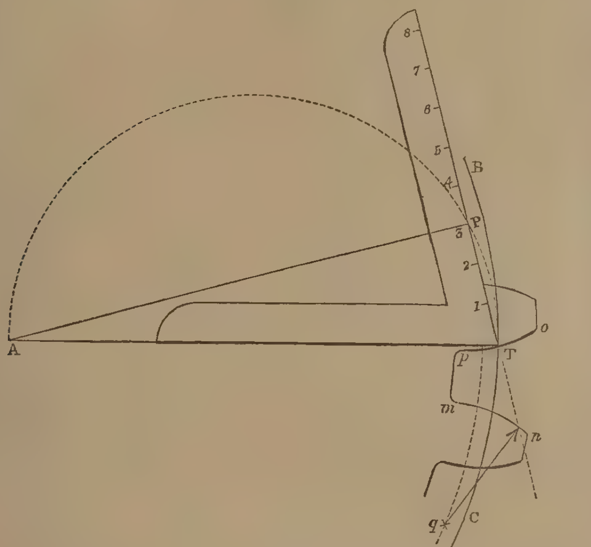
* When pattern shapes, or *moulds*, as they are commonly termed, are employed to lay in the teeth by, the pattern ought to contain two teeth and a space between; or better, one complete tooth to draw by, and a half tooth by which to set the mould. The half tooth ought to be accurately cut in the direction of its length to the radius of the wheel, so that being applied to the radial lines drawn on the wheel pattern, through the pitch divisions on the pitch circle, the arc of that circle drawn on the mould, shall exactly coincide with an equal arc of it drawn on the wheel. If still greater attention to accuracy be desired, the mould may be made with three pattern teeth, and tested by observing that the outline of the succeeding, always corresponds to the draught of the preceding tooth.

but the abstruse manner in which he treated the subject seemingly caused his labours to be overlooked until the subject was finally taken up by Professor Willis, and placed within the comprehension of every one of ordinary mathematical acquirements.

But before entering upon the geometrical consideration of the conditions involved, it may simplify the investigation to explain, in the first place, the practical application which Mr. Willis has made of it in the construction of his Odontagraph, an instrument which deserves to be extensively known and understood. The fundamental principle which it involves may be thus exhibited:—

Let A be the centre and AT the radius of the pitch circle of a proposed wheel; draw TP, making an angle ATP=75°30′, with the radius, and drop a perpendicular AP upon TP. Or draw AT, and upon it describe a semicircle, and cut off TP=¼ of AT, that is equal to a quarter of the radius; then will P be the centre from which, if an arc *op* be described through T, that arc will be the side of the tooth required.

Fig. 50.



This principle being established, it may be adapted conveniently to practice by constructing a bevil of brass, of the requisite form, the angle at T being 75½ degrees: the side TP may for convenience be graduated into a scale of quarters of inches as in the figure; and these divisions may be further subdivided if thought proper. If the bevil so formed be laid upon the radius AT of the proposed wheel and its point T to the pitch circle, the centre point P will be found at once by reading off the length of the radius of the wheel in inches upon the reduced scale. Thus in the figure the radius AT is three inches long, and the point P is found at 3 of the scale.

To draw the curves of the other teeth, describe from centre A with radius AP the circle *pq*, dotted in the figure, and falling within the pitch circle; this will be the locus of the centres of the teeth. Thus having divided the pitch circle into arcs corresponding to the pitch of the teeth, take the constant radius PT in the compasses, and keeping one point in the dotted circle as at *q*, step from tooth to tooth describing successively the arcs of the faces first to the right and then to the left. For example, *mn* is described from centre *q*, and *po* from P.

The sides of teeth thus formed, consisting each of a single arc, bear considerable analogy to involute teeth, and like these have the fault of acting together with rather too much obliquity; but the mode of describing them is exceedingly simple, and they have this advantage that they will work correctly with wheels of any number of teeth, with any amount of clearance, and without any great nicety of adjustment of the pitch circles. For wheels of a small pitch —as ¾ inch and under—this mode of setting out may be adopted without risk; but when the sides of the teeth consist of single arcs, there can only be one position of action in which the angular velocity will be strictly constant —and that is when the point of contact is on the line of centres. This objection is obviated by making the side of each tooth to consist of *two arcs*, joined at the pitch line, and so struck that the point of action of the one shall be a little before the line of centres—say at the distance of half the pitch; and the exact point of the other at the same distance beyond that line.

The mode of drawing these arcs is very fully described by Professor Willis, but it is much too complex to be rendered available in the workshop; besides it has been rendered superfluous for practical purposes by his Odontagraph, in which it is embodied. This instrument, which is fast coming into use, is usually formed of card-paper, sheet-brass, or plane-tree. Fig. 52, (see page 112), is a representation of it exactly two-thirds the original size as constructed by Messrs. Holtzapffel and Co. of London, and contains within it the requisite tables of numbers.

One side is graduated into a scale of half inches, each half inch being again subdivided into ten equal parts. The line *t*D, which corresponds to the radius of the wheel, makes an angle of 75° with the side. The mode of using the instrument is thus explained in its application to the setting out of a wheel of 29 teeth, of 3 inch pitch.

Describe an arc of the required pitch circle, and upon it set off the pitch T*t*, fig. 51; bisect this in *e*, and draw radial

Fig. 51.

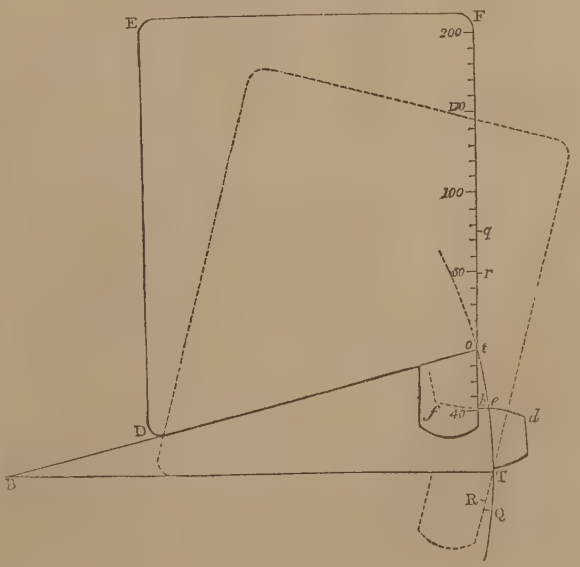
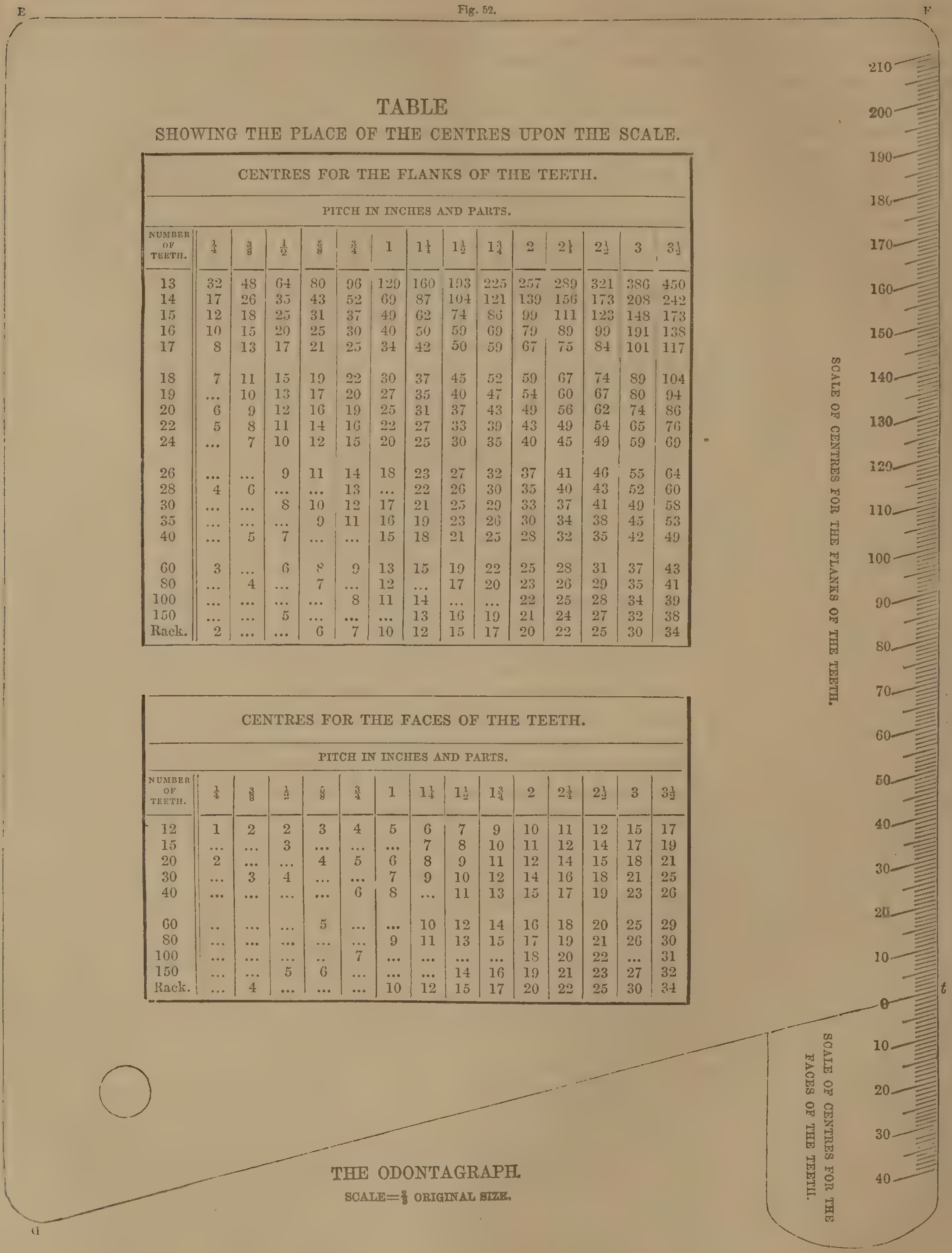


Fig. 52.


E
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SCALE OF CENTRES FOR THE FLANKS OF THE TEETH.
SCALE OF CENTRES FOR THE FACES OF THE TEETH.

TABLE
SHOWING THE PLACE OF THE CENTRES UPON THE SCALE.

CENTRES FOR THE FLANKS OF THE TEETH.

NUMBER OF TEETH.	PITCH IN INCHES AND PARTS.													
	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	3	3½
13	32	48	64	80	96	129	160	193	225	257	289	321	386	450
14	17	26	35	43	52	69	87	104	121	139	156	173	208	242
15	12	18	25	31	37	49	62	74	86	99	111	123	148	173
16	10	15	20	25	30	40	50	59	69	79	89	99	191	138
17	8	13	17	21	25	34	42	50	59	67	75	84	101	117
18	7	11	15	19	22	30	37	45	52	59	67	74	89	104
19	...	10	13	17	20	27	35	40	47	54	60	67	80	94
20	6	9	12	16	19	25	31	37	43	49	56	62	74	86
22	5	8	11	14	16	22	27	33	39	43	49	54	65	76
24	...	7	10	12	15	20	25	30	35	40	45	49	59	69
26	...	...	9	11	14	18	23	27	32	37	41	46	55	64
28	4	6	...	...	13	...	22	26	30	35	40	43	52	60
30	...	...	8	10	12	17	21	25	29	33	37	41	49	58
35	...	...	...	9	11	16	19	23	26	30	34	38	45	53
40	...	5	7	...	...	15	18	21	25	28	32	35	42	49
60	3	...	6	8	9	13	15	19	22	25	28	31	37	43
80	...	4	...	7	...	12	...	17	20	23	26	29	35	41
100	...	...	...	...	8	11	14	...	...	22	25	28	34	39
150	...	...	5	...	...	...	13	16	19	21	24	27	32	38
Rack.	2	...	...	6	7	10	12	15	17	20	22	25	30	34

CENTRES FOR THE FACES OF THE TEETH.

NUMBER OF TEETH.	PITCH IN INCHES AND PARTS.													
	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	3	3½
12	1	2	2	3	4	5	6	7	9	10	11	12	15	17
15	...	...	3	...	...	...	7	8	10	11	12	14	17	19
20	2	...	...	4	5	6	8	9	11	12	14	15	18	21
30	...	3	4	...	...	7	9	10	12	14	16	18	21	25
40	...	...	...	...	6	8	...	11	13	15	17	19	23	26
60	..	...	...	5	...	...	10	12	14	16	18	20	25	29
80	...	...	...	...	...	9	11	13	15	17	19	21	26	30
100	...	...	...	..	7	...	...	...	...	18	20	22	...	31
150	...	...	5	6	...	...	...	14	16	19	21	23	27	32
Rack.	...	4	...	...	...	10	12	15	17	20	22	25	30	34

THE ODONTAGRAPH.
SCALE=⅔ ORIGINAL SIZE.

lines BT and B*t*. For the arc within the pitch circle apply the slant edge of the scale to the radial line B*t*, placing its extremity *t* on the pitch circle as in the figure. In the table headed *Centres for the Flanks of the Teeth*, look down the column of 3-inch pitch, and opposite to 30 teeth—which is the nearest number to that required—will be found the number 49. The point *r* indicated on the drawing board by the position of this number on the scale of parts marked "*Scale of centres of Teeth* within the *Pitch Circle*," is the centre required, from which the arc *ef* must be drawn with a radius *re*.

The centre for the arc *de*, which lies outside the pitch, is found in a manner precisely similar, by applying the slant edge of the scale to the radial line BT. The number 21 obtained from the table of *Centres* for *the Faces of the Teeth*, will indicate the position of this centre upon the scale of *Centres for Teeth outside the Pitch Circle*, namely, at R.

The double curve *df* is also true for an annular wheel of the same number of teeth, *f* becoming of course the point of the tooth and *d* its root. For a *rack* the pitch line T*t* will be a straight line, and B*t*, BT, drawn perpendicular to it, at a distance from each other equal to the pitch.

The numbers for pitches not stated in the table, may be obtained from the column of some other pitch by direct proportion. Thus for 4-inch pitch, by doubling the number of the column of the 2-inch pitch; for 4½ by doubling 2¼, and so on; or if the difference be small, the column belonging to the nearest pitch may be employed without a serious error; or more accurately, a number may be taken half-way between those given in the two nearest columns. Also no tabular numbers are given for 12 teeth in the upper table, because within the pitch circle their teeth are radial lines.

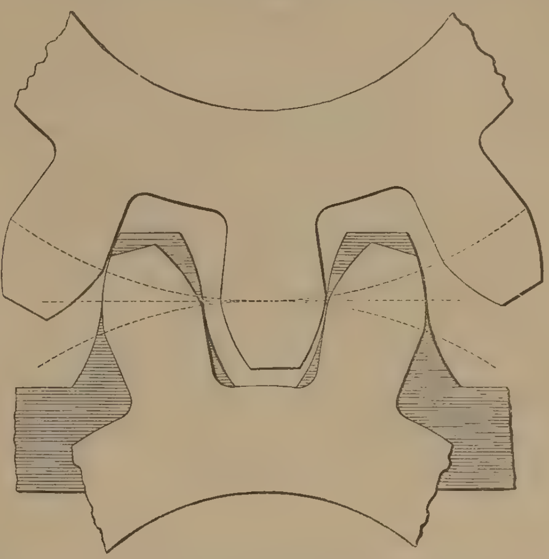
In reference to the numbers given in the tables, the inventor observes that it is unnecessary to extend these to every wheel, as the error produced by taking those belonging to the nearest, as above directed, is so small as to be unappreciable in practice. He has calculated the amount and nature of the error by way of obtaining a principle for the number and arrangement of the wheels selected, and without entering upon a detail of the processes of calculation we may briefly state the results.

The difference in form between the tooth of one wheel and of another is due to two causes: the first is the difference of curvature, which is provided for in the Odontagraph, by placing the compasses at the different points of the scale of equal parts; and secondly, the form depends on the variation of the angle TB*t* (fig. 51,) which is met by placing the instrument upon the two radii in succession.

The first is the only cause with which these calculations are concerned. Now in three-inch pitch, the greatest difference of form produced by mere curvature in the portion of tooth which lies beyond the pitch circle, is only $\frac{1}{25}$ of an inch between the extreme cases of a pinion of 12 teeth and of a rack; and in the acting part of the arc within the pitch circle it is $\frac{1}{10}$ of an inch; so that as all the other forms lie between those two, it is clear that if we select only four or five examples for the outer side of the tooth, and ten or twelve for the inner side, we can never incur a greater error than the $\frac{1}{250}$ part of an inch, in a three-inch pitch, by always taking the nearest number in the manner directed, and a proportionally smaller error in smaller pitches. But to insure this, the numbers ought to be so taken, that the corresponding form shall lie equally between the extremes. This is more necessary with wheels of small numbers of teeth, since the variation of form is much greater among the teeth of low than among those of larger number.

Fig. 53, annexed, shows the relation of the curves of the teeth of a rack and pinion of 12 teeth, both adapted to work into the same wheel. They are drawn with accuracy, and on a large scale, so as to afford a graphical comparison of the two extreme cases.

Fig. 53.



The Odontagraph is likewise applied to determine the correct form for *cutters* used in the wheel-cutting machine, for shaping the teeth of metal and cog wheels. The form of the cutter is manifestly that of the space between two contiguous teeth; and may therefore be determined by drawing a pair of teeth in any particular case. But in making a set of cutters especially for small pitches, it is not by any means necessary to make one for every number. The forms for numbers of teeth that lie closely together are, as already remarked, very nearly alike—so nearly alike indeed, that the errors of workmanship would entirely destroy the difference, particularly when the numbers are high. When the numbers are very low, there is less room for deviation from the strict rule—not more, for example, between the form of a cutter for 16 and 17, than there is for a cutter for a wheel of 150 teeth and another for one of 300. This being the case, it appeared to Mr. Willis to be worth while to investigate some rule by which the necessary cutters could be determined for a set of wheels, so as to incur the least chance of error; and to this effect he has calculated, by a method sufficiently accurate for the purpose, the following series of what may be termed equidistant values of cutters; that is, a table of cutters so arranged that the same difference of form exists between any two consecutive numbers.

TABLE of EQUIDISTANT VALUES for CUTTERS.

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
No. of Teeth.	Rack.	300	150	100	76	60	50	43	38	34	30	27	25	23	21	20	19	17	16	X	15	14	13	X	12

This table affords the requisite information in the selection of the wheel to which each cutter shall be accurately adapted, after it has been determined how many are necessary in a set. For example, if a single cutter were thought sufficient for a set of very small wheels, it had better be accurately adapted to teeth of 25, for that value is intermediate between the two extremes. If three cutters are to suffice for the whole set, then 76, 25 and 15 ought to be selected—of which the cutter 76 may be used for all teeth from a rack to 38; the cutter 25 from 38 to 19, and the cutter 15 from 19 to 12; and so on. In cutters, the greatest difference of form is at the apex of the tooth, that is, at the base of the cutter, and this amounts to $\frac{1}{4}$ of an inch in 2 inch pitch, when the teeth have the usual addendum. From this the difference for any smaller pitch may be ascertained, and as many cutters interposed as the workman's notion of his own powers of accuracy may induce him to think necessary. Thus, if the hundredth of an inch be his limit of accuracy in forming his cutters, and he proposes to make a set for half inch pitch, where the difference of form is $\frac{1}{4} \times \frac{1}{4} = \frac{1}{16}$, that is, $\frac{6}{100}$ nearly, then half a dozen cutters will be sufficient, and these must be made as nearly as possible to suit the wheels 150, 50, 30, 21, 16, 13.

The following table contains a selection of numbers for different cases which may save trouble:—

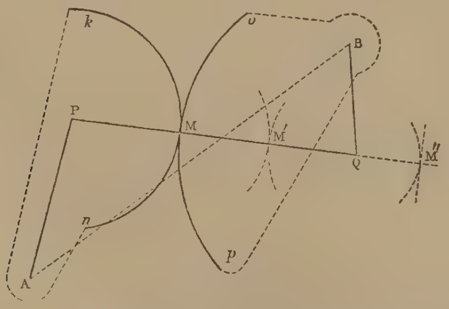
TABLE of CUTTERS.

No. of Cutters in the set	Numbers of Teeth to be selected.																							
2	50	16																						
3	75	25	15																					
4	100	34	20	14																				
6	150	50	30	21	16	13																		
8	200	67	40	29	22	18	15	13																
10	200	77	50	35	27	22	19	16	14	13														
12	300	100	60	43	34	27	23	20	17	15	14	13												
18	300	150	100	70	50	40	30	26	24	22	20	18	16	15	14	13	12							
24	Rack.	300	150	100	76	60	50	43	38	34	30	27	25	23	21	20	19	18	17	16	15	14	13	12

When the numbers have been selected, the Odontagraph may be employed to draw the figure of the cutter corresponding to each wheel, either on the same scale, or better, on a larger scale, which may be afterwards reduced proportionally to the size required.

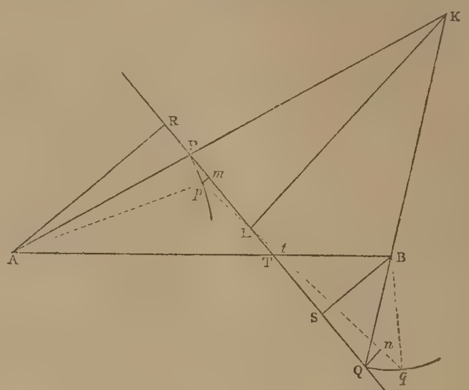
To explain the principle of the Odontagraph, let us, in the first place, observe the action of the two pieces in the annexed diagram, of which P and Q are the centres

Fig. 54.



of curvature at the points of contact, and which are capable of revolving round A and B. It is plain that the line PQ is, in every position of contact of these curves, equal to the sum of their radii, and must therefore be constant during the motion of the segments through the small arcs of constant curvature. It is therefore manifest that if, for the segmental pieces, a pair of radial arms, AP and BQ, connected by a link PQ, be substituted, their angular velocities will have the same ratio as those of the segments which they represent. And it is also equally clear, that so long as the *ratio* of the lengths remains constant, any difference of the actual lengths of the arms will not affect the motion ratio. Let then the rod AP be moved into a new position A*p*, as shown in fig. 55; its extremity will manifestly carry with it the end of the link PQ, and communicate through it a motion to the arm BQ, causing it to assume likewise a new position B*q*. It is necessary to know the relative value of this motion to that of AP which produced it, which though constantly changing may be thus determined at any instant:

Fig. 55.



When the rod PQ changes its position, it may be considered during its motion to turn about some centre in space, although the relative position of that centre be perpetually shifting. This centre must of necessity be the point in which the arms AP, BQ would intersect if produced. Thus the momentary centre, when the motion begins, is K. For as the extremity P moves round the centre A, the direction of its motion at starting must be perpendicular to AP, and therefore the momentary centre must be somewhere in the line AP produced. In like manner the initial motion of the extremity Q must be perpendicular to BQ, and the momentary centre must lie somewhere in the direction BQ: but these directions intersect in K, which is therefore the centre in space about which the rod PQ on starting into motion may be conceived to turn. And since the rod PQ turns about K, the direction of motion of P and Q are to each other at any given instant as their radial distances from K; that is, as PK and QK, which is true whether we consider them as the extremities of the rod PQ or of the radii AP, BQ; also, the angular motions of the latter will be found by dividing those direct motions by their radii; that is,

Angular motion of

P round A: angular motion of Q round B :: $\frac{PK}{AP} : \frac{QK}{BQ}$

If now we draw KL, AR, BS, perpendicular to PQ,

PK : AP :: KL : AR by similar triangles KPL; APR
BQ : QK :: BS : KL - - - - - BQS ; KLQ
AT : BT :: AR : BS - - - - - ART; BTS

And compounding these three proportions we obtain

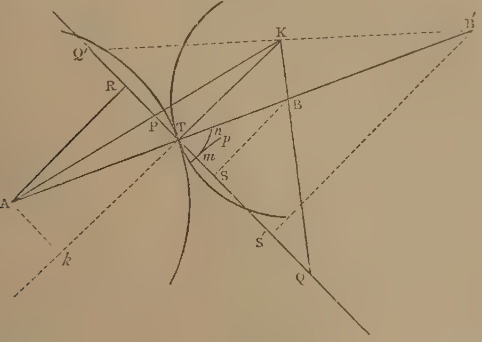
$$\frac{\mathrm{PK}}{\mathrm{AP}} : \frac{\mathrm{QK}}{\mathrm{BQ}} : : \mathrm{BT} : \mathrm{AT}$$

that is to say, the angular motions of the arms are to each other at any moment inversely as the segments into which the link divides the line AB joining the centres of motion, or *line of centres*. If now the link move into a new position *pq* near the first, and it happen that this second position intersects the first in a point L above or below the line of centres, as shown in the figure, then the ratio AT : BT will be changed into A*t* : B*t*, and consequently the angular motion will be an increasing or decreasing ratio. But if the point L coincide with the line of centres, this ratio for the moment will remain constant.

Now, a little consideration will show that the point of intersection between two successive positions PQ, *pq* of the link must be at the place where the perpendicular from K falls upon it. For as K is the momentary centre of motion, the extremity L of the perpendicular will begin to move in a line at right angles with it, and consequently will remain in the direction of the first position when the link has passed into the second position; in other words, it will be in the point of intersection of the two positions. When, therefore, the rods are in such a position that the perpendicular from K meets the link PQ in the line of centres, the ratio of the angular motions of AP and BQ is constant; and if in this state the points P and Q be employed as centres whence short arcs are drawn through any common point M′, as in fig. 54, and applied as teeth, these arcs will manifestly drive each other correctly when in the exact relative position described, and very nearly so when removed to a short distance on either side of it, which is the thing required.

Now these relative positions of P and Q may be determined by a simple construction founded upon the necessary coincidence of L and T. Thus let A and B in fig. 56 be the centres of a pair of wheels, and AB the line of centres divided at T, the point of contact of the pitch circles. Draw PTQ, making any angle with the line of centres, and upon it assume P as a centre from whence the circular side is to be described of a tooth of the wheel whose centre of motion is A. To find the corresponding centre for a tooth of the wheel which turns upon B, draw TK perpendicular to PQ, produce AP to meet it in K; join KB and produce it to meet PQ in Q, then will Q be the required centre—as will appear by comparing this figure with the preceding. If then a small arc *mn* struck from P be taken as the face of a tooth of the wheel on A, it will work correctly with an arc *mp* struck from Q through *m*, and employed as a tooth of the wheel on B.

Fig. 56.



Were BQ so taken as to make the angle KBT acute—for example at B′, then would Q fall at Q′ on the same side of T as P, but beyond it, and the effect would be that the tooth *mp* would be concave instead of convex. And if the angle KBT = PTA, then will KB become parallel to PT, and the point Q being thus removed to an infinite distance, the arc *mp* of the tooth of the wheel on B will be a right line perpendicular to PT.

The angle ATP is arbitrary, and its value may, therefore, be determined from other conditions than those stated. It may be remarked, however, that if made a right angle, the points P and Q will vanish by coinciding with T; and if it be made a little less than a right angle, the points P and Q will be thrown so near to T, that the radii by which the arcs are struck become too short, and the points of the teeth too much rounded off. On the other hand, if the angle ATP be made too acute, the action of the teeth upon each other at the moment of passing the line of centres and elsewhere becomes very oblique, and an injurious pressure is thereby thrown upon the axis of the wheels. By various trials Mr. Willis has adopted 75° as the value of the angle which appears to avoid these extremes, and he has accordingly employed it in the construction of his Odontagraph.

Again, the position of the point *m*, through which the arcs are to be struck, is also arbitrary, and must be determined by considering which point of the action we wish to make the correct point. If the teeth consist of a single arc each, the correct point may be fixed at the moment of passing the line of centres, and therefore the arcs must be struck through the point T; but if the side of the tooth be formed of two arcs joined, one lying within and the other beyond the pitch line, then the action of one of them will be confined to the approach of the point of contact of the teeth to the line of centres, and the action of the other to its recess from that line, and *m* must be assumed upon such a principle that the correct point of each arc shall fall nearly in the middle of its action.

The distance between the centres may also be found by calculation, thus: AR being perpendicular to PT, we have by the similar triangles ARP; PTK

$$\mathrm{KT} = \frac{\mathrm{PT} \times \mathrm{AR}}{\mathrm{PR}} = \frac{\mathrm{PT} \times \mathrm{AR}}{\mathrm{TR} - \mathrm{PT}}$$

$$= \frac{\mathrm{PT} \,.\, \mathrm{AT} \,.\, \sin \mathrm{ATP}}{\mathrm{AT} \,.\, \cos \mathrm{ATP} - \mathrm{PT}}$$

$$\text{therefore } \mathrm{PT} = \frac{\mathrm{AT} \,.\, \mathrm{KT} \,.\, \cos \mathrm{ATP}}{\mathrm{KT} + \mathrm{AT} \,.\, \sin \mathrm{ATP}} = \mathrm{D}$$

which is the centre for the arc *mn*. And similarly drawing BS perpendicular to TQ we have for the corresponding arc *mp*.

$$\mathrm{QT} = \frac{\mathrm{BT} \,.\, \mathrm{KT} \,.\, \cos \mathrm{ATP}}{\mathrm{KT} + \mathrm{BT} \ \sin \mathrm{ATP}} = d$$

Now the point P being obtained from the point K on the opposite side of T to A, these formulæ belong to that part of the tooth of the lower wheel beyond the pitch line and the flank of the tooth of the upper wheel. The corresponding face of the upper and flank of the lower wheel may be determined in a similar manner, by joining B*k* and following the course of reasoning stated for the point K.

In these formulæ, the values of PT and QT may be determined for any forms of teeth, and also for a set of wheels which shall work correctly together. But the greatest value which in this last case can be correctly assigned to KT must be one which, if r be the smallest radius, shall make

$$\text{KT} = r \sin \text{ATP}$$

Consequently putting R for the radius of the wheel, and angle ATP $= \theta$, the values of D and d, that is, the distances of the centre points of the arcs measured from T, will become

$$\text{D} = \frac{\text{R} \,.\, r \,.\, \cos\theta}{\text{R} + r} \qquad d = \frac{\text{R} \,.\, r \,.\, \cos\theta}{\text{R} - r}$$

By assuming constant values of r and θ in a set of wheels, the values of D and d which correspond to different numbers and pitches may be calculated and arranged in tables for use. In this way the tables which accompany the diagram of the Odontagraph in p. 112, were obtained by taking the least number of teeth to be given to a wheel $= 12$ and $\theta = 75°$, the numbers being expressed in twentieths of an inch.

To illustrate this calculation by an example, let the radius of the wheel be 20 inches, and let it be one of a set of which the least is to have 12 teeth; further, let the given pitch be 3 inches;

$$\text{then } r = \frac{p.n}{2\,\pi} = \frac{3 \times 12}{2 \times 3.1416} = 5.73 \text{ inches};$$

p being the given pitch $= 3$ inches and n the number of teeth.

$$\text{Also, } \cos\theta = \cos 75° = .25882$$

$$\text{Hence D} = \frac{20 \times 5.73 \times .25882}{20 + 5.73} = 1.16 \text{ inch.}$$

Now, a wheel of 20 inches radius and 3 inch pitch will have 42 teeth, the curves of the faces of which must be struck with a radius of 1.16 inch; or expressed in twentieths by a radius of 23.2, which agrees with the number for 40 teeth found in the lower table of the Odontagraph.

$$\text{Similarly } d = \frac{20 \times 5.73 \times .25882}{20 - 5.73} = 2.08 \text{ inch nearly,}$$

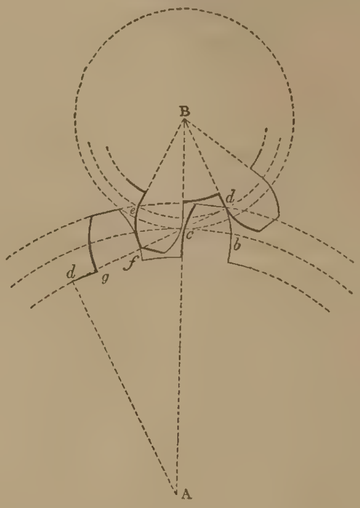
and this expressed in twentieths is 41.6, which corresponds to the number in the upper table.

In this way these tables may be extended at pleasure, and for any values of r and θ; and perhaps it would be preferable in the construction of such tables to express the numbers in inches and 32nds of an inch, which would better adapt them to the common foot-rule.

Involute teeth.—Involute teeth have the disadvantage, already stated, of being, when in contact, too much inclined to the radius, by which an undue pressure is transferred to their axes. Their mutual friction is thereby little affected, but that of the axes is increased, and their journals are more speedily worn. But they have at the same time the advantage of working with more accuracy under derangement and incorrectness of fitting, and any pairs of them will work truly together in sets within certain limits, however different in diameters, the pitch being the same. Until Professor Airy had developed the mode of adapting the epicycloid to the condition in question, the involute was the only form known in practice by which the conditions of perfect figure for wheels of any sizes to work smoothly in wheels of any other sizes, could be satisfied.

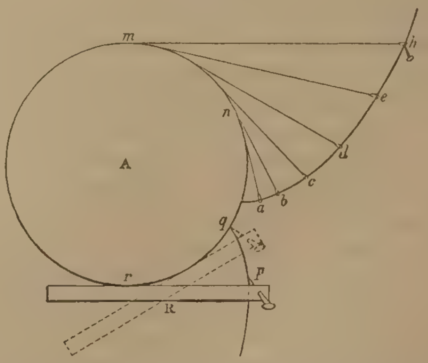
To describe this curve for the teeth of a pair, of which the radii of the pitch circles, and pitch of the teeth are determined, we may employ the mode illustrated by the annexed figure. Let A and B be the centres of the pair, and *e*, *b* be their pitch lines; join A and B by a right line passing through *c;* from this last point draw *cd*, *cd* perpendicular to the radials B*d*, A*d*, and cutting them in *d* and *d;* this line *dd* is then a common normal to the teeth in contact, and the perpendiculars A*d*, B*d*, are the radii of the involute circles which form the acting faces of the teeth.

Fig. 57.



The involute curve may be described mechanically in several ways. Thus let A in the annexed figure be the centre of a wheel, for which the form of involute teeth is to be found. Let *mna* be a thread lapped round its circumference, having a loop-knot at its extremity *a;* in this fix a pin, with which describe the curve or *involute* *ab* - - - *h*, by unwinding the thread gradually from the circumference, and this curve will be the proper form for the teeth of a wheel of the given diameter.

Fig. 58.



The involute *ab* - - - *h* may also be produced by an epicycloidal motion; for since the circumference of a generating circle, whose centre is infinitely distant, must be a straight line, we may form the involute by making a straight ruler roll upon the circumference of the circle to be evolved. Thus let R in the preceding figure be a straight ruler, at whose extremity a pin p is fixed with the point resting upon the point *q* of the circle; then by rolling the straight ruler upon the circumference, so that the point at which it touches the circle may move gradually from *q* towards *r*, the curve *qp* will be generated exactly similar to the involute *ab*----*h*, and the method is perhaps easier and more accurate than by unlapping a thread from the circumference of the *evolute*, as above described.

Mr. Hawkins, in his Appendix to Camus' Treatise on the Teeth of Wheels, recommends a very simple instrument for striking involute teeth. Two views of this instrument are given in the annexed figures. In these *ab* is a piece of watch-spring, with two teeth *c*, *c*, formed upon its edges; at *a* is a small screw by which the spring is attached to a templet formed of thin board, and equal to a sector of the circle upon which the involute is intended to be generated. At *b* a small bit of wire is inserted into the spring, to form two knobs as a handle by which the spring can be kept stretched and moved laterally. The templet is equal in the thickness to the width of the spring.

Fig. 59.

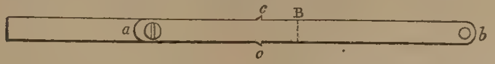
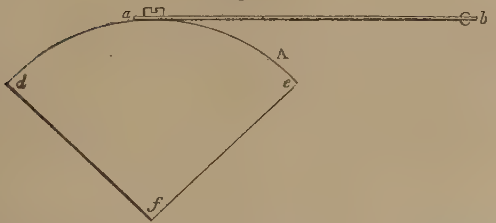


Fig. 60.



In using the instrument, the templet is to be fixed upon the drawing board or surface upon which the involute is to be traced; the centre *f* coinciding with the axis of the wheel, or its representative on the drawing board; then by holding the knobs *b* as a handle, and keeping the spring thereby tightly stretched, so that it remains a correct tangent to the circle, and lapping the spring upon the arc of the templet, one of the points *c* will trace upon the surface over which it passes a curve which, when the point has arrived at the circumference of the pitch circle represented by the templet, will be a true involute of that circle.

If the tracing instrument be now turned over, the opposite tooth *c* on the side of the spring now undermost, will by repeating the same operation, trace the counter involute for the other side of the tooth of the wheel.

Mr. Hawkins' mode of proceeding to draw the outlines of a pair of involute wheels with his tracer is simple, and may be explained by reference to fig. 57. Having drawn the line of centres AB, and divided it at *c* into parts A*c*, B*c*, proportionate to the intended velocities of the two wheels, next draw the right line *dd* cutting the line of centres at *c*, and making with it an angle deviating not more than 20 degrees from a right angle. From centre A describe a circle touching the line *dd* at *d* within the pitch circle of that wheel, and which will thus have *dd* as a tangent at *d*. In like manner from B as a centre describe a circle touching the same straight line at *d* within the pitch circle of that wheel, and also having that straight line as a tangent. Then the distance between the two circles is the proper length of the teeth, from which, however, the necessary clearance ought to be deducted. The two circles last drawn are likewise the bases upon which the involutes for forming the boundaries of the teeth are to be generated.

Next, having provided two involute tracers made in the way above described, each with a templet corresponding to a sector of the base circle of its wheel, the centre *f* of the templet is to be applied to the centre A or B of the wheel of the pair to which it belongs, and *d* to the point *d*; then supposing the templet for A to be first used, let it be adjusted until the tracing point *c* of the spring coincides with some point as *d* of the base circle of the wheel B, by moving it until it meets the circumference of its own base circle at *b*, the side *db* of the tooth *dcb* will be formed, and of the proper length, by moving the templet spring over the space included between the exterior of one base circle and the other. By turning over the templet, and adjusting it for the required thickness, the outline of the opposite side of the tooth may be traced in the same manner. Repetition of the operation with the templet of B will give the form of the teeth of that wheel.

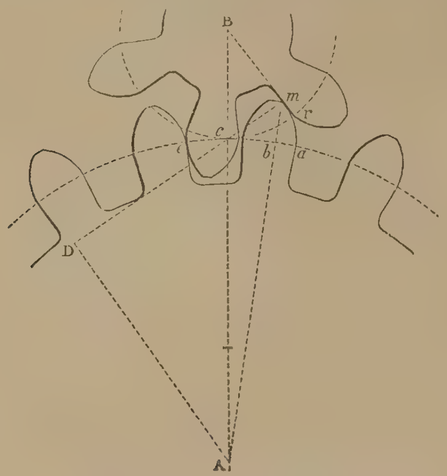
If thought desirable all the teeth of the pair may be drawn in in the way indicated, or a single tooth of each wheel may be accurately drawn, and two pattern teeth cut to the forms thus found.

Mr. Hawkins states that teeth formed according to this method will communicate equable motion from either wheel to the other, without possessing any tendency to press the shafts outwards—which tendency would exist if the angle of the common tangent (*dd*) of the two base circles deviated much more than twenty degrees from a right angle with the line of centres.

If the teeth of both wheels are to be of equal thickness at the base, which they ought to be when of the same material for the sake of equal strength, the part *gc* of the common tangent ought to be bisected in the middle *f* of its length, and the involutes drawn in the way described will give the roots of the teeth so nearly equal, that a very slight correction only will be required—the amount depending on the relative diameters of the wheels. But when the teeth of one wheel are required to be thicker than those of the other, the part *gc* of the common normal *dd* must be unequally bisected in proportion to the required difference.

It is finally to be observed that the teeth may have their bases in any other circles than those given, since the radii of the generating circles are entirely arbitrary; but the proportion taken is found to satisfy the conditions required with very little outward action. In respect to the character of the involute curve as compared with the epicycloid, it is obvious that the former possesses the advantage that a greater number of teeth of equal strength may be given to a wheel by this than the epicycloidal form; for with the latter the space must be at least equal to the tooth, while not much more than half the space is required to be marked off on the base circle for an involute tooth of the same length. There are also more teeth engaged at the same time, and thus the strain is divided.

Fig. 61.



Teeth of a Rack and Pinion.—A rack, as before observed, may be regarded as a wheel of infinite diameter. The teeth may, therefore, be defined by the epicycloidal method explained for spur wheels, by making their flanks straight, and the teeth of the pinion involutes. Now, the properties which have been shown to belong to involute teeth manifestly obtain, however great may be the dimensions of the pitch circle of their wheels, or whatever disproportion may exist between them. Thus of the two wheels A and B with involute teeth, which work together, let the radius of the pitch circle A*c* of one become infinite; its circumference will then become a straight line represented by the face of a rack. But whilst the radius A*c* becomes infinite, the radius AD of the circle from which its involute teeth are struck, and which bears a constant ratio to the first, will also become infinite, and therefore the involute *ma* will become a straight line, perpendicular to the pitch line at *a*. For it is evident that the extremity of a line of infinite length, unwinding itself from the circumference of a circle of infinite diameter, will describe through a finite space a straight line perpendicular to the circumference of the circle. The involute teeth of the rack will, therefore, have their faces perpendicular to the pitch line; and this line will be determined by drawing a tangent to the circle on B through the point of contact from which the involute teeth of the pinion are to be struck. If the radius B*m*, with which the *involute* circle of the pinion is described, be equal to the radius of its *pitch* circle, the line D*m* will become parallel to the face of the rack, and the sides of the teeth of the rack perpendicular to it. But in this modification, the teeth of one wheel (B) remain unaltered, and the accuracy of their action uninfluenced by the change in the dimensions of the pitch circle of the other, which has converted it into a rack, and its curved into straight teeth; it therefore follows that straight teeth upon a rack must necessarily work truly with involute teeth upon a pinion.

But as straight teeth are not the form of greatest strength which may be attained, and as strength is of importance as well as uniformity of action, the conditions of the problem may be examined with a view to the best modification which the form admits with respect to this test. If with the pitch radius B*c* of the pinion we describe the pitch circle, and draw MN a tangent to it at *c*, we next make this right line the pitch line of the rack; and, further, describe the circle whose radius is B*h*, making this last the base of an involute *hb*; then if the tooth of the rack be bounded by a right line *h*DE, making an angle BDM with the pitch line equal to B*ch*, and the involute *hb* be moved into the position *h'b'*, it will drive the sloped tooth to the position P*m*, always touching it in the line *hc*H; and the velocity of the circumference of the pitch circle will always be equal to that of the pitch line, since

$$\mathrm{DP} = \frac{h\mathrm{V}}{\sin h\mathrm{D}c}$$

Since also $h\mathrm{V} = \text{arc } hh'$, by the property of the involute

$$= \text{arc } pp' \times \frac{\mathrm{B}h}{\mathrm{B}c}$$

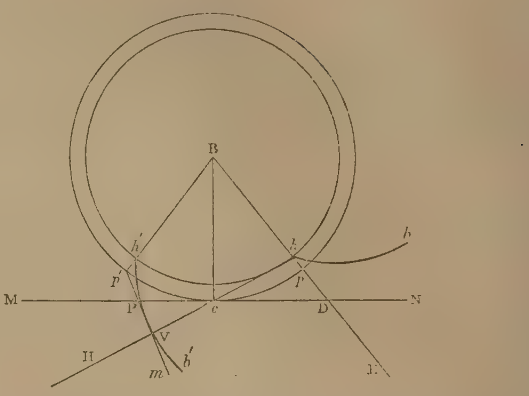
$$= \text{arc } pp' \times \sin \mathrm{B}ch$$

$$= \text{arc } pp' \times \sin h\mathrm{D}c$$

And therefore DP = arc *pp'*. Hence a wheel with involute teeth will work truly with a rack whose teeth are straight-sided, and inclined to the pitch line at an angle θ, provided only

$$\frac{\text{radius of base}}{\text{radius of pitch circle}} = \frac{\mathrm{B}h}{\mathrm{B}c} = \sin\theta$$

Fig. 62.



In a rack of this kind, the locus of contact of the teeth being the tangent line *h*H, the sloping teeth will be pressed downwards by a certain portion of the working pressure, which in some cases may be an advantage in itself by neutralising vibration, and the advantage of a greatly stronger form of the teeth is at the same time secured.

The teeth of the pinion being of the regular epicycloidal form, the faces of the teeth of the rack will have a cycloidal form. But according to the usual method of setting out racks and pinions, the pitch line of the rack is the locus of contact, and the action on one side of the line of centres is confined to a constant point in the rack tooth, which is thus subjected to rapid wear. This disadvantage may be abridged by reducing the length of the teeth of the pinion, but it may also be entirely avoided by taking a constant generating circle, and employing it to describe cycloidal flanks for the rack teeth by rolling on its pitch line, and then by describing the faces of the teeth of the pinion with the *same* generating circle, in the manner before described. By this simple modification, the contact will no longer be confined to the pitch line of the rack, but will be distributed over the flanks of its teeth; and these, it may be observed, may be made of any length desired, by regulating the diameter of the generating circle accordingly. If the generating circle have a diameter equal to the radius of the pinion, the flanks of the pinion-teeth will then be radial, and this form may also be modified by taking a circle of greater or less diameter, as explained at p. 110.

To determine the pitch circle of the pinion, let D represent the distance through which the rack is to be moved by each tooth of the pinion, and let these teeth be in number = N; then will the rack be moved through the space N×D during one complete revolution of the pinion. Now the rack and pinion are to be driven by the action of their teeth, as they would by the contact of the circumference of the pitch circle of the pinion with the plane face of the rack, so that the space moved through by the rack during one complete revolution of the pinion must precisely equal the circumference of the revolving pitch circle; consequently, calling R the radius of the pitch circle of the pinion, we will have

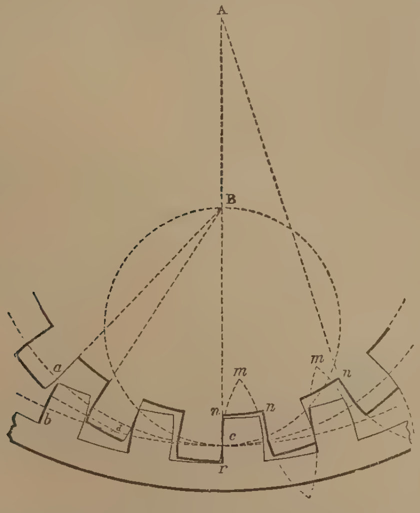
$$2\pi R = N \times D \quad \therefore R = \frac{N \times D}{2\pi}$$

Two of these quantities being given, the other immediately becomes known. Thus let D = 2 inches, and N = 13, then

$$R = \frac{26}{2 \times 3.1416} = 4.138 \text{ inches.}$$

Annular Wheel and Pinion.—An annular wheel is one having its teeth formed within its periphery, and consequently the pinion works internally. In this arrangement, the form of the teeth may be determined according to the principles already explained, with very slight modification in their practical application.

Fig. 63.



Thus in fig. 63, let A, B, be the centres of the wheel and pinion; and let c be the point of contact of the pitch circles. The epicycloid cm described by the circle having a diameter $= Bc$, gives the lateral form of the tooth of the wheel, limited as before by a radius drawn to the middle of the thickness of the tooth; and an arc described from the centre A, with radius An, cuts the diameter Bc at a point which determines the side of the tooth of the pinion at n. But if we attempt to apply the same process to finding the face of the tooth of the pinion formerly employed in determining the flank of the wheel-teeth, we shall find that the construction will either not give the side of the tooth as the arcs do not meet, or it will give it a position upon the radius Ac on the same side as the face cm. From this, then, it appears that the teeth of the wheel will have no flanks, and that their outline will consist simply of *faces*. But then it follows that the *pinion* would not drive the wheel uniformly according to the required condition, and in consequence we must find another curve for the faces of the teeth of this last. Now if we trace the epicycloid by a point c of the circle to centre A in the figure, rolling it externally upon the circumference of the circle on B, it will give the outline of a tooth capable of fulfilling the condition of uniformity of action at any point of the circumference of the circle on A. Should the teeth engage before the line of centres, the tooth of the wheel would necessarily act constantly at that point and cause great wear, particularly as this species of geering is usually employed in conveying the motion of water-wheels where the acting surfaces are always wet.

In practice it is found sufficiently accurate to substitute for the epicycloidal method above described for finding the form of the face of the tooth of the pinion, the more convenient mode of defining it by an arc of a circle drawn from the root of the tooth, with a radius equal to the chord of the pitch.

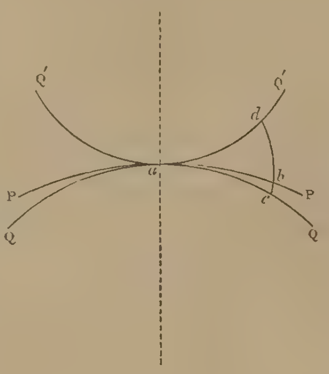
When the teeth do not engage until they arrive at the line of centres, the faces of the teeth of the pinion may be entirely suppressed, leaving the flanks only, which are alone acted upon by the teeth of the wheel. This may be uniformly done, at least when the pitch of the pinion is small, and when the wheels are more than strong enough for the power which they are required to transmit. In such cases, the following method of setting out the teeth of the pair may be adopted with safety.

In fig. 63, let A be the centre of the wheel, and B the centre of the pinion as before. From the point c of contact of the two pitch circles mark off the arcs ca, cb, equal respectively to the pitch of the pinion and wheel; join aB and ba, and bisect this last; from the point of bisection raise a perpendicular, and the point d, where this meets the pitch circle of the wheel, will be the centre from which the arc ab, defining the side of the tooth of the wheel, may be struck. The flank of the tooth of the pinion is determined by the two radii which limit its thickness and meet in B. To define the length of the teeth, it is sufficient to describe a circumference with the radii Ar, Bn, leaving sufficient bottom clearance; and the extremities of the teeth of the pinion, and the bottoms of those of the wheel, ought to be rounded off by arcs drawn with a radius equal to the chord of the pitch taken upon the pitch circle of the pinion.

By this method it will be seen that the strength of the pinion-teeth is greatly reduced, and ought therefore only to be applied when the power to be transmitted is comparatively light, and when the wheel drives the pinion. It is consequently not recommended for general practice. The epicycloidal method, explained for external wheels, may, on the contrary, be applied in all cases, observing that in the case considered, the *space* answers to the tooth, and conversely the tooth to the space; so that keeping this distinction in view, the forms of the teeth may be determined with as much readiness, as if the pair were of the ordinary kind of spur geer. The teeth may also be set out with the Odontagraph, if that instrument be in use in the pattern shop.

There is one property of the internal pinion which may be here pointed out, namely, the diminished friction which attends its working, as compared with an external pinion of the same diameter. To render this obvious, let PP, in the annexed figure, be the pitch circle of the wheel, QQ the pitch circle of an internal pinion, and Q′Q′ that of an external pinion of equal diameter. Then supposing the wheel to drive, and that it moves until the point a arrives at b; by this advance, the same point of the pinions will have arrived respectively at d and c of the circles QQ and Q′Q′, having both moved over a space equal to ab. Now it is manifest that the distance from b to c is less than the distance from b to d; but these distances may be assumed as the measure of the amount of slide of the teeth of the respective pinions in moving to their new position; for whatever may be the actual amount of slide, it must evidently be in the ratio of bc to bd, since these quantities measure the velocity of the sliding contact of the pair. But again, when the circumstances are otherwise identical, the friction is as the space passed over. It is, therefore, evident that the internal pinion has less friction than the external one; and the same figure may be employed to show that it has a larger arc of contact with the wheel, and is, therefore, other things being equal, stronger than the external pinion.

Fig.64.

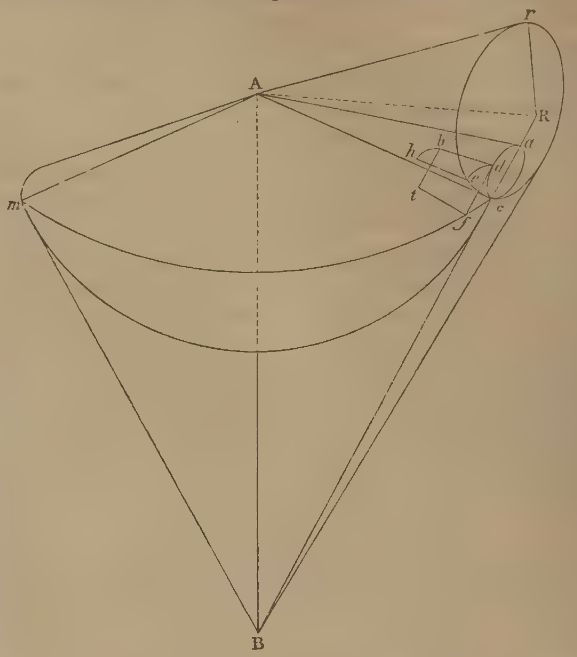


Teeth of Bevil Wheels.—When a cone is made to revolve upon the surface of another fixed cone, in such a manner that their summits always coincide, the curve which is generated by a point in the circumference of the rolling cone is a kind of epicycloid, which will plainly lie on the surface of a sphere whose centre is the common apex of the cones, and whose radius is equal to the slant side of the rolling cone. From this property the curve is termed the *spherical epicycloid;* and if the cone roll on the concave surface of the base, the curve produced is denominated the *spherical hypocycloid.*

To apply this definition, let there be two cones in contact along their slant sides Ac; and let cAa be any other cone, having its line of contact coinciding with Ac, and having its apex at A; the axes of these three cones will then be in the same plane ARB. The circumferences will also be at the same distance Ac from A, and will lie on the surface of a sphere whose centre is A and radius Ac.

Now, suppose the three cones to revolve about their respective axes with the same relative velocity as would be produced by the rolling contact of their surfaces, it is clear that the line of contact will constantly be Ac, and calling aAc the *describing cone*, a line db upon the circumference of the surface of this directed to the common apex A, will generate one surface $fbdt$ on the outside of the cone Acm, and another surface $dehb$ on the inside of the cone cAr. These surfaces will touch along the describing line db; for since $ftdb$ is generated by the rolling of the generating cone on the cone mAc, the motion of db is at every instant perpendicular to the line of contact Ac, and therefore the normal plane at db to the surface generated by that line, must pass through the line of contact Ac. In like manner the normal plane to the surface $hedb$ will pass through the line of contact of the cones, and consequently the surfaces must touch along db.

Fig. 65.

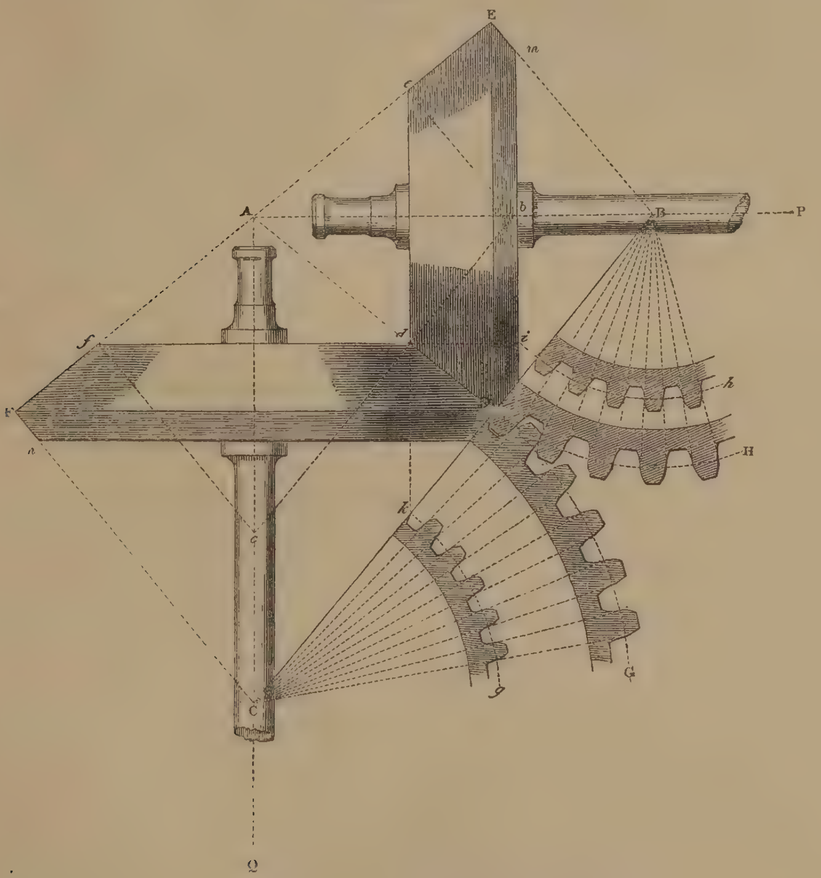


Now, under these conditions, it is clear that if the rotative motion of the cone mAc be communicated by teeth to the cone cAr by their contact action, the teeth being generated by the describing cone aAc, the motion will be the same as that produced by the rolling contact of the conical surfaces, for at the beginning of the motion ft and eh coincide with the line of contact Ac; and since the arcs cf, ce, described by the bases of the cones, are, respectively, equal to cd, and therefore to each other.

In practice the portion of spherical surfaces occupied by these arcs, when employed as teeth, are narrow belts of the cones extending only a small distance within the circle of the bases, and therefore cones tangent to the sphere along cm and cr may be substituted for the sphere itself without practical error. Thus draw RcB perpendicular to the line of contact Ac, and intersecting the axes of the two cones at B and R; then BR revolving about AB will generate a conical surface tangent to the sphere along the base cm; and the same line RB revolving about AR will generate a conical surface tangent to the sphere along the base cr of the cone cAr. And since the arc df, which really lies in the spherical surface, is very short, it may be supposed, without sensible error, to lie in the surface of the tangent cone *c*B*m*, and to be described with a circle whose diameter is equal to that of the base of the describing cone. And, in like manner, the arc *de* may be supposed to be similarly described upon the surface of the tangent cone *c*A*r*.

These principles furnish us with a ready mode of obtaining the form of the teeth of a pair of bevil wheels. Thus let AP and AQ be the axes of the shafts between which the motion is to be transmitted. The angular velocities with which the two shafts are driven being determined, draw AD such that it divides the angle PAQ into two parts, proportional to the numbers of teeth in the wheels respectively, or inversely proportional to the angular velocities or numbers of turns per minute of the wheels on the shafts. If we conceive the line AD to revolve about the axis AP, at the same angle PAD, the conical surface ADE will evidently be described, a portion of which, DE*ed*, is represented in the figure. Similarly, by the revolution of the line AD about the axis AQ, the conical frustum DF*fd* will be described. It is obvious, therefore, that these two cones, communicating the motion by simple contact, will transmit motion in the given ratio, and hence they are termed the *primitive cones*, and have an obvious analogy to the primitive circumferences or pitch circles of spur wheels. In the interior of their mass the flanks and spaces are conceived to be formed, and upon their surfaces are placed the projecting faces of the teeth—the returning surfaces E*m* and F*n* of the cones, or the *crowns*, as they are termed, are defined by inverted conical surfaces formed upon the bases DE and DF; they are short frusta of the cones BDE and CDF, the two apices of which are determined by the line BDC drawn at right angles to AD. To perceive the propriety of this form of the crown, it is only necessary to consider that the ends of the teeth E*m* and F*n* will thus always be square with the surface of the primitive cones, and they will geer most completely into each other. The other ends of the teeth are defined in like manner by the conical recesses *ebd* and *dcf*, which are parallel to the exterior surfaces of the crowns; the line *bdc* being, like the exterior line, at right angles to AD.

Fig. 66.



It appears, then, that the true forms of the teeth may be described with facility upon the surfaces of the cones DEB and DFC; and for our present purpose of representing the forms of these teeth upon a plane surface, it is only necessary to conceive those surfaces spread out or developed upon a plane. Now, the development of a cone, that is, the form assumed by the superficies when laid out flat, is a sector of a circle, of which the radius is equal to the slant height of the cone, and the arc equal to the circumference of the cone at the base. In the case before us, therefore, if a circular arc DG be described from the centre C with the radius CD, it will be the development of a portion of the base, and the sector CDG will likewise be the development of a portion of the cone; and by describing an arc *kg* with the radius $Ck = cd$, the annular space contained between kg and DG will be the development of the face of the wheel. It is upon this annular segment, then, that the forms of the teeth may be drawn precisely as if it were a part of a spur wheel; and if we suppose a piece of sheet copper or other suitable material to be cut into the form of the teeth, and wrapped upon the crown of the wheel, the outline which could thus be traced off would represent the true forms of the teeth of the bevil wheel.

The breadth D*d* being settled by the same rules as for spur wheels, it is obvious that the teeth must follow the taper of the cone towards the point A. Draw *dk* parallel to AC, and from the centre C describe the arc *kg*; then, as before, the segment C*kg* will be the development of a portion of the cone *dcf*, and upon this segment the forms of the upper ends of the teeth may be described. Lines drawn from the teeth on the arc DG to the centre C, will determine the magnitude of the teeth on the arc *kg*, and the teeth may thence be described, as in the figure.

The largest diameter DF is reckoned the diameter of the wheel, and, similarly, DE is said to be that of the pinion. The process which we have detailed for describing the teeth of the wheel is precisely the same as that for describing the teeth of the pinion. It is unnecessary, therefore, to particularize further, than that the figure shows the teeth of the pinion drawn similarly to those of the wheel, upon the arcs DH and *ih*.

That the teeth thus formed will work truly together, becomes obvious, when we reflect that if any two exceedingly thin wheels, with the form of teeth described, had been taken in a plane perpendicular to AD passing through the point D and having their centres in the axes of the given wheels, they would work truly together, and their angular velocities would be in the given ratio. Now it is evident that the portion of each of the conical surfaces which is at any instant passing the line *ki* is at that instant revolving in the plane perpendicular to AD which passes through the point D, the one surface revolving in that plane about the axis AC, and the other about the axis AB. Those portions of the teeth of the wheels which lie in these two conical surfaces will, therefore, drive one another truly, at the instant when they are passing through the line *ki*, if they be cut of the form which they must have had, in order that they might drive one another truly and with the required angular velocity, had they acted entirely in the plane perpendicular to AD and about the given axes. But this is precisely the forms in which the teeth are supposed to be cut, and therefore those portions of the bevil teeth which lie in the conical surfaces will drive one another truly at the instant they are passing through the line *ki*; and, therefore, they will drive one another truly through a small arc on either side of that line, which is the condition required, since it is only through an exceedingly small distance on either side of that line that any two given teeth remain in contact. It therefore follows, that those portions of the teeth which lie in the conical surfaces D*f*, D*e*, work truly with one another throughout the whole breadth of the conical belt.

If the radius of the base of the frustum be called R, and the radius of its developed circle be r; also, if the semi-angle CAD of the rolling cone $= \theta$; then $r = \frac{R}{\cos\theta}$. Hence the action of the teeth in any bevil wheel is equal to that of a spur wheel whose radius $= \frac{R}{\cos\theta}$; and if N be the number of teeth in the bevil wheel, $\frac{N}{\cos\theta}$ will be the number of those in the spur wheel of equal action. Thus if $\theta = 50°$, then $\frac{R}{\cos\theta} = \frac{1}{.64279} = 1.556$, and therefore the action of a bevil wheel of 50° is fully equivalent to that of a spur wheel having a radius one-half greater, and consequently a half more teeth.

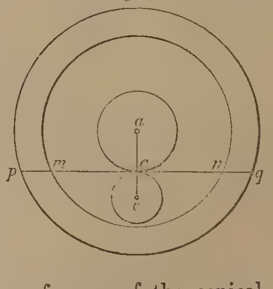
This, then, is a reason for the superior action of bevil wheels, as compared with spur wheels of the same number of teeth; for spur wheels always act the better the more teeth they have, and it appears that the bevil wheel is always equivalent in its action to a spur wheel of a greater radius, and consequently a greater number of teeth.

When a pair of bevil wheels have equal numbers of teeth, and have their axes at right angles to each other, they are termed *meter wheels.* In this case $\theta = 45°$ and $\cos\theta = .70711$; therefore taking radius equal to unity, we have $\frac{1}{\cos\theta} = 1.4$; in words, the action of a meter wheel is nearly equivalent to that of a spur wheel, with half as many teeth.

Skew Bevils.—When the axes are inclined to each other, and yet do not meet in direction, and it is proposed to connect them by a single pair of bevils, the teeth must be inclined to the base of the frusta to allow them to come into contact.

To find the line of contact upon a given frustum of the tangent-cone; let the annexed figure be the plane of the frustum; *a* the centre. Set off *ae* equal to the shortest distance between the axes (called the *eccentricity*,) and divide it in *c*, so that *ac* is to *ec* as the mean radius of the frustum to the mean radius of that with which it is to work; draw *cp* perpendicular to *ae*, and meeting the circumference of the conical surface at *m*; perform a similar operation on the base of the frustum by drawing a line parallel to *cm* and at the same distance *ac* from the centre, meeting the circumference in *p*.

Fig. 67.



The line *pc* is then plainly the line of direction of the teeth. We are also at liberty to employ the equally inclined line *cq* in the opposite direction, observing only that, in laying out the two wheels, the pair of directions be taken, of which the inclinations correspond.

The annexed figure, which we here repeat for the sake of convenient reference, renders this mode of laying off the outlines of the wheels at once obvious. In this figure, the line *ae* corresponds to the line marked by the same letters in the preceding diagram; and the division of it at *c* is determined in the manner directed. The line *cm* being thus found in direction, it is drawn indefinitely to *d*. Parallel to this line and from the point *e* draw *e* to *e*, and in this line take the centre of the second wheel. The line *cmd* gives the direction of the teeth; and if from the centre *a* with radius *ac* a circle be described, the direction of any tooth of the wheel will be a tangent to it as at *c*. And similarly if a centre *e* be taken in the line *ed*, and with radius $ed = ce$ a circle be drawn, the direction of the teeth of the second wheel will be tangents to this last, as at *d*.

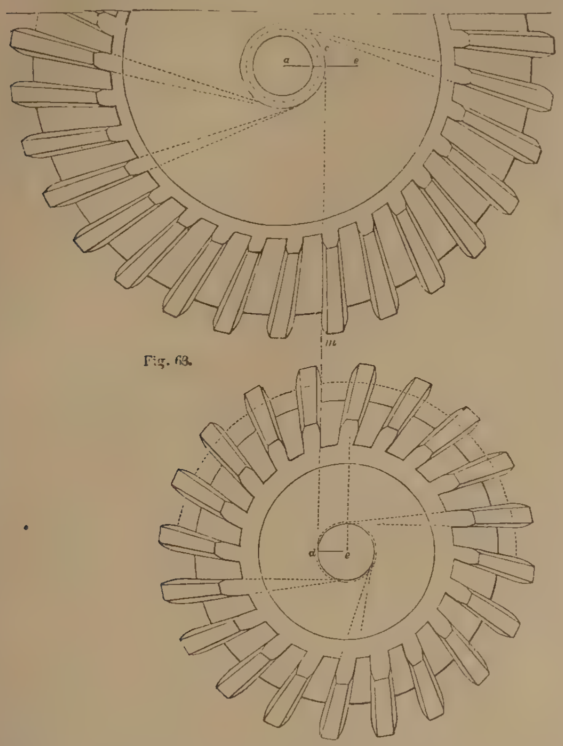


Fig. 68.

Having thus found the direction of the teeth, their outlines may be described exactly as in the case of ordinary bevil wheels, and with equal exactness and facility.

Wheel and Tangent Screw.—This is often a very convenient and ready mode of reducing a high velocity.

To determine the form of the teeth of the wheel and thread of the screw, it may be remarked that, from the nature of the screw, the section of its thread, made by a plane passing through its axis, is everywhere the same; and that if a series of such sections of the entire screw be made by planes at equal angular distances round the circle, a set of figures, resembling a double rack, will be obtained alike in the number and form of their teeth, but in which the teeth will approach nearer and nearer to the extremity of the screw. Now, while revolving, the screw successively brings these sections into operation upon the whole teeth, producing exactly the same effect as if the screw were pushed end-long without rotation, in the manner of a rack. And this supposition furnishes a ready mode of obtaining the forms of the wheel-teeth and thread of the screw, upon the principle of the rack and pinion.

In the annexed figures *ac* is the line of centres of the wheel and screw. The screw is shown in section by a plane cutting it in the line of its axis. Now, the screw being considered a driving rack, it passes a tooth of the wheel during each revolution, and, therefore, the point *a* will, at the end of a revolution of the screw, be the point *b*; but the thread must necessarily have continued in contact with the tooth while passing from *a* to *b*. Now, if it be required that the thread of the screw shall be in contact with the tooth at a point only, the tooth will be straight, and its obliquity equal to the pitch of the screw. But if it be desired that the thread shall be in contact with the entire side of the tooth, the outline of the wheel-teeth must be different in every section perpendicular to the axis of the wheel. The required form will be found by making such series of sections of the screw as proposed, and adapting the portion of the side of the tooth to that particular section with which it is intended to work. And since in every section two or even three teeth may be in simultaneous contact, the screw may be in contact along the entire side of those teeth.

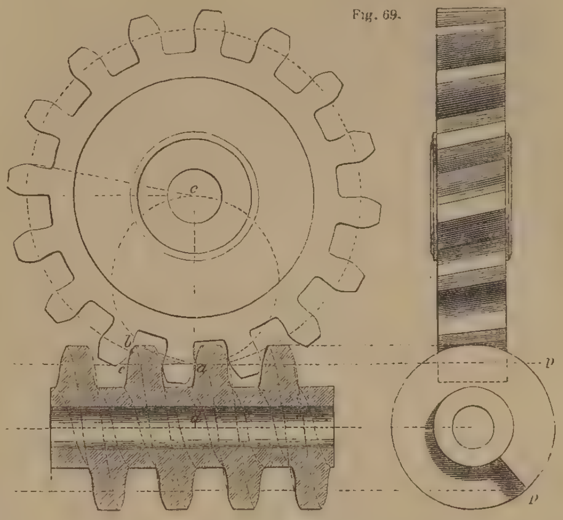


Fig. 69.

In practice, then, the form of the tooth must be determined for the respective sections of the screw, as in the case of the rack; and the thread of the screw may obviously take the same sloped form if great strength be required.

To find the diameter of a wheel, driven by a tangent-screw, which is required to make one revolution for a given number of turns of the screw, it is obvious, in the first place, that when the screw is single-threaded, the number of teeth in the wheel must be equal to the number of turns of the screw. Consequently, the pitch being also given, the radius of the wheel will be found by multiplying the pitch by the number of turns of the screw during one turn of the wheel, and dividing the product by 6.28.

SECTION VII.—ON THE CONSTRUCTION OF WHEEL-PATTERNS.

When a wheel-pattern is to be made, the first consideration is the determination of the diameter to suit the required speed; the next is the pitch which the teeth ought to have so that the wheel may be in accordance with the power which it is intended to transmit, and for which a safe rule is given in page 93. These being determined, the number of the teeth in relation to the pitch and diameter will be found from the table and rules in page 83. The proportions of the teeth, the clearance and length, will be found by reference to the tables and diagrams of pages 89—91, and in page 88 is explained the principle on which the breadth of the wheel is defined in relation to the length of the teeth.

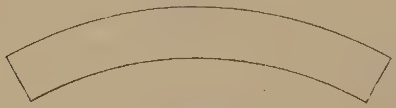
The size and proportions of the wheel being thus settled, the operation of constructing the pattern is ready to be undertaken. Let us, in the first place, assume that the wheel is to be a radial one with six *arms;* and here it must be premised, that the number of arms increases according to the diameter of the wheel, thus, with occasional deviation:

Wheels from 1½ to 3¼ feet diameter have 4 arms.
" from 3¼ to 5 feet diameter have 5 "
" from 5 to 8¼ feet diameter have 6 "
" from 8¼ to 16 feet diameter have 8 "
" from 16 to 24 feet diameter have 10 "

It is assumed in the numbers of this table, that the wheels are cast complete, the boss supporting the arms, and these last the rim with the teeth cast upon it. The cases of deviation are those in which the wheels are very small, and cast with a continuous *web*, instead of radial arms, and where the wheels are required to be particularly light, and to possess great accuracy: it is in this case preferable to increase the number of arms, so that they may be made thinner, and thereby incur less risk of the rim being drawn out of form during the cooling of the metal.

In arm-wheels the rim is always built of segmental pieces, cut out of deal of convenient thickness with the bow-saw, to an external radius equal to the pitch radius of the wheel, which allows a depth equal to the flanks of the teeth for dressing. A templet or *mould* is, in the first place, made of thin, well seasoned and dry deal, and by this the form of the building segments is drawn upon the wood from which they are to be cut. Suppose that the rim is of such breadth that six thicknesses of wood are required to make it up; then four segments are cut of the requisite breadth and to the same internal radius, and two of them to an internal radius less than the former by at least as much as the intended thickness of the rim. With these segments the rim is to be built in the following manner:

Fig. 70.



Segment Templet.

A circular plate of wood, with an iron centre on the back of it, usually with an internal screwed boss on the centre of it, having the same pitch of thread as the thread of the screw on the projecting end of the lathe-spindle, is prepared and turned up truly on the face. A circle is also described upon it to indicate the pitch line of the wheel, and to this circle the exterior arcs of the rough segments are to be adjusted.

Thus prepared, the first course of segments are planed true on one side, and fitted with sprigs upon the plate, their ends accurately jointed together, glued, and sprigged. When the glue is in some degree *set*, the plate is put upon the spindle of the lathe, and the segmental ring is turned true on the flat side; but the interior and exterior circumferences are left in their rough state. The plate is then removed from the lathe-spindle—usually a simple spindle with not more than two speeds upon it. Another course of the segments is now dressed with the plane on one side and built upon the first. This course is bedded with glue and sprigged to the first course, observing that the joints of this course do not coincide with those of the first, but that all the former joints are covered by the whole wood of the last laid course. The glue being allowed time to set, the plate is again put upon the lathe-spindle, and the face of the ring is dressed as before, still leaving the circumferences in their rough state.

The segments of these two courses are of the same breadth, but the next course to be laid must have the increased breadth necessary to form the *web* or *feather* of the rim; this, as already stated, is usually made equal in breadth to the thickness of the rim. This course will, therefore, project inwards over the ring formed by the two courses already laid. In other respects, however, it is not different, and it is laid and fastened in exactly the same manner as the second course.

Two courses are laid of these broader segments; afterwards two others of segments of the same breadth as the two courses first laid. This done, the building of the rim is complete, and a cross section of it would present the appearance shown by the part *adbc* of fig. 72 and an arc of it in elevation is represented by the part A*b*B*a* of fig. 71. In fig. 72, the courses are numbered in the order in which they are supposed to be laid, the layers 3 and 4 being those which are intended to form the web of the rim.

Fig. 71.

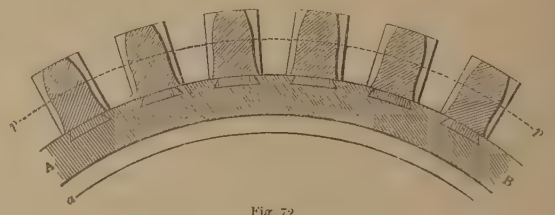
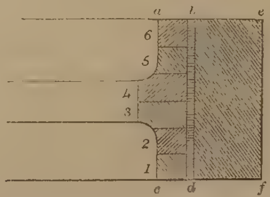


Fig 72.



This process being completed, and the face-plate placed on the lathe-spindle, the exterior edge of the rim and half the interior circumference, that is, the interior of the three courses marked 4, 5, 6, are turned true to the required form. The rim is then taken off the face-plate and reversed; but is now fastened to the plate by screws passed from the back of the plate into the edge of the rim.* In this position the interior of the other half of the breadth of the rim is dressed, that is, the inner circumferences of the layers 1, 2, 3, when the rim is ready to have the arms fitted into it.

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* Or, a better mode of fixing the rim, in so far as it does not injure it by holes, is to reverse it exactly concentric with the face-plate, this meantime being laid level on the bench or on trestles, and to fix it in

In being dressed, the web or feather formed by the two middle layers 3 and 4 is diminished in thickness, till at their edge they are together exactly of the same thickness as the arms which are to be put into the pattern; and a good practical rule is to make these equal in thickness to *seven-eighths* of that of the rim of the wheel. In breadth they ought to be equal at least to the breadth of the rim, when this is calculated according to the rule of maximum strength already pointed out, and towards the centre should be increased to $\frac{7}{6}$ths of that breadth. More commonly they are made equal in breadth to the rim, and tapered to $\frac{5}{6}$ths of that breadth at their extremities.

The breadth of the arms being determined, they are to be fitted into the rim by a *half-check* upon the web, which is done by checking out at six points, answering to the number of the arms, the breadth of an arm in the web layer marked 3. The extremities of the arms are then checked to the same depth and fitted into their places with a little glue, and two short but strong screw-nails at each end.

The mode of fitting the arms together is different in different shops; but the easiest mode is to fit a double arm in the manner described, that is, an arm extending through the diameter, and having a half-check at the middle of its length cut to the proper angle to receive the second double arm, extending like the first, through the complete diameter. These being fitted in, two half-length arms are then inserted, butting into the wide angles left at the centre between the arms already fitted. Fillets are next inserted to fill up and round off the acute angles, both at the centre and extremities of the arms; these are intended to give additional strength to the wheel, and to facilitate the extraction of the pattern from the sand in the process of moulding. Were the acute angles left open, the sand would be ready to break down and give the wheel an uncouth and rough appearance, a result which is to be guarded against in all kinds of moulding, and more especially in wheels.

The flat arms being thus fitted in, the next operation is to insert a centre, upon which the boss or eye of the pattern may be at any time built according to the size necessary for the particular wheel to be cast off the pattern. The centre is formed of two hexagons, one laid on each side, and built of six separate pieces, or made of single solid blocks, of half the depth of the rim, so that on each side the centre projects beyond the plane of the rim by half the thickness of the flat arms. The centre is firmly screwed together, and to the arms of the pattern; and on one side—the side which is intended to be uppermost in the process of moulding—a flat plate of iron, usually of the same form as the centre itself, is indented and fastened with screws to the wood. This plate has a hole in the centre, intended to admit an iron prong for starting and lifting the pattern out of the sand.

The next operation is to attach the feathers of the arms. These are placed along the middle of the flat arms, the planes of the two being at right angles to each other, so that their cross section at any point is represented by +. The breadth of the feathers diminishes from the *centre* towards their extremities, in exactly the same ratio as the flat arms; at the centre they are half the width of the rim, and consequently stand *flush* with the depth of the centre, but at the extremities which meet the rim the width is reduced a sixth. Fillets are fitted in to fill up the angles which the ends thus form with the inside of the rim, for the reasons already stated. Small *running* fillets are likewise fitted into the angles which they form with the flat arms.

We have described the insertion of the centres and arm feathers as if the operations were performed simultaneously on the two sides of the pattern. This is hardly correct; for in the actual operation, the centre and feathers are commonly inserted while one half of the inside of the rim is in its rough state, and while it is still attached to the lathe plate. One side being finished, the pattern is taken off and reversed in the manner described, to allow of the second half of the interior of the rim being dressed; and without removing the pattern from the plate, the remaining centre and arm-feathers are fitted in as before.

The interior of the pattern may now be considered as finished; and it may be remarked that it is always desirable to have the arms completely fitted and secure, before commencing to turn the exterior of the rim; for should the rim be dressed on the outside previously to the arms being put in—which is practised by some makers—it is hardly possible to prevent its springing, and being thrown out of truth. By the process described, the arms are fitted in while the exterior of the rim is in its rough state, and consequently although it should slightly spring, the external and most important circumference will be brought to the truth in the after process of turning. The rim is also stronger and less liable to be thrown from the truth, while in the rough state, than it is after being finished to the thickness.

After the arms are fitted into the interior of the wheel, and finished in the manner described, it is taken from the face-plate; a hardwood plate, with a centre for the lathe-spindle, is screwed upon the eye of the pattern; it is then placed in the lathe, and the exterior of the rim is turned to the required thickness. In this process it receives a very slight taper—usually from an eighth to a tenth of an inch in a foot—to allow it to draw cleanly from the sand in the process of moulding. About the same fraction is allowed upon the diameter of the pattern for contraction of the metal in cooling.

The next part of the operation is to cut the dovetail seats of the teeth. The form of these recesses is shown in fig. 71, beneath the teeth. They extend completely across the breadth of the rim, and are intended to receive the dovetail pieces (*bd* fig. 72) on which the teeth are fixed. The dovetail pieces are themselves made of hardwood, and besides being otherwise convenient, they serve to bind the several layers of the rim firmly together.

These pieces are, in the first place, accurately fitted into the seats and marked, so that every piece may afterwards be

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this position by four or more blocks nailed upon the face-plate, and at the same time sprigged to the outside of the rim. These sprigs do comparatively little injury to the soft wood of the rim, into which they seldom penetrate deeper than the part subsequently to be dressed off.

inserted into the proper groove to which it was fitted; the rough pieces of which the teeth are to be formed—these being cleaned with the plane on one side—are then fitted, each with a dovetail fastened by screws to its dressed side; and being cut nearly to the proper length, they are fitted into their places by the dovetails, which are made with a slight taper, so that they can be the more easily driven in its seat, and taken out again when required. The blocked teeth being thus all placed and partially fixed upon the circumference of the pattern, this is again placed on the lathe for the purpose of dressing off the ends and points of the teeth. This done, the pitch circle of the wheel is finally marked off upon the ends of the teeth with a fine point.

The pattern is now ready to have the form of the teeth drawn in, as represented in fig. 71, by an arc of the rim, at this stage, with the outlines of the teeth also drawn. In this figure the pitch line is denoted by the dotted circle *pp*, and the form of the teeth is denoted by the shaded portions of the rectangular blocks. The various methods of obtaining the outline have already been fully explained.

To finish the pattern, it therefore only remains to dress off the teeth to the forms indicated, and to fix the dovetails permanently in their beds. This last is easily effected by a film of glue; and if the teeth are to be finished by the wheel-cutting machine, it ought to be performed previous to placing the pattern in the lathe for the purpose of dressing the ends and points of the teeth, and consequently before pitching them. But when the teeth are to be finished by hand, the order of the operation is that described and the most commonly practised.

In finishing the teeth by hand, they are removed in succession, by driving out the dovetails, one only at a time; the unshaded portions of the block are then accurately dressed off by concave and convex gauge-planes adapted to the purpose; the dovetail is then fitted into its bed with a film of glue; and the tooth is complete. Every tooth is treated in exactly the same manner until the whole number be dressed and fastened upon the circumference of the rim. But unless the operation of fixing the teeth be performed with care, there is danger of twisting the rim and throwing it out of truth. To guard against this, the safest practice is to take always in succession the pair of teeth opposite to each other on the circumference: that is, having fitted a tooth on one side of the pattern, the tooth opposite to it on the other side, ought to be the next taken, so that any tendency of the glueing to twist the rim at one point will be neutralized by an opposite tendency in fitting the next tooth.

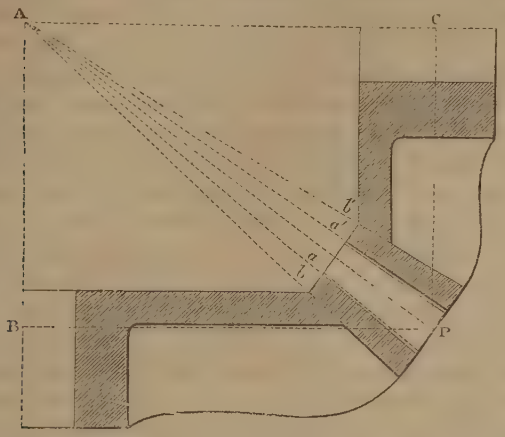
The wheel pattern may now be considered complete. There are many minute details of a technical kind which could only be appreciated by a workman who has a knowledge of the operations. These it has not been thought necessary to notice, as they would suggest themselves to any intelligent workman, and therefore would serve no useful purpose in a general description of the order of the operations, which is the object here aimed at. It may also be remarked that certain proportions have been laid down as empirical rules to be followed in the construction of the pattern, without assigning any reasons for adopting these proportions rather than others. Within certain narrow limits these proportions may undoubtedly be departed from without injury to the wheel; the arms, for example, may be made broader, the rim slightly thicker, and the feathers deeper, without any bad effect. The purpose to be attained by any set of proportions which can be suggested is simply, that the greatest amount of strength be secured with the least amount of material under the conditions imposed by the diameter of the wheel. Now, it is obvious that the wheel will be no stronger than its weakest part; and supposing the thickness of tooth determined, it is clear that the rim does not require to be stronger than the teeth it carries; they ought, therefore, to be proportional to each other, according to the conditions under which they severally support the strain of the power to be transmitted by the wheel. And if the rim and teeth ought to be proportional to each other, and of a given strength, it is clear that this will also be the amount and limit of the strength of the arms, since these need not manifestly be stronger than the rim, which is equal in strength to the teeth. Now, this equality of strength is nearly attained by the proportions stated, which is the condition required to be fulfilled.

It is also to be remarked, that in making the pattern of a cast-iron wheel, it is necessary to take into account the nature of the material; the pattern must not only be of such a proportion of parts as to be sufficiently strong, calculated by the series of the parts; it must also be so proportioned that the fluid metal, when poured into the mould, shall *set* in every part as nearly as possible at the same instant. For if the parts contain disproportionate quantities of metal, it is plain that the thinner parts will cool more quickly than the others, and as cooling is attended with contraction, the parts of the casting will be put upon strain from their contracting irregularly and faster at one point than at another; and if the irregularity be carried to a certain extent—and the limits are not wide—the casting, on being removed from the mould, will be found fractured through some of the thinner and first cooled and first contracted parts. Thus, one of the most common errors is to put too much thickness into the boss of the wheel, and in consequence it not unfrequently happens that the arms are drawn away from the rim. To avoid this, the metal round the eye ought never to be more, in arm-wheels, than twice the thickness of the rim; that is, equal to the pitch of the teeth, and this is quite sufficient in strength to resist the driving of the keys in fitting the wheel on the shaft, provided this operation be done with ordinary skill and care.

To prevent the bad effects of unequal contraction, the arms are sometimes made of a curved form. When the wheel is very light in proportion to its diameter, the arms are advantageously made with a double curve like the old letter ʃ, which admits extension to a certain extent, depending on the elasticity of the material, in the direction of its length, without fracture. Pulley-arms are very often and very judiciously made of this form.

The rules and directions which apply to spur wheels, likewise apply, with slight technical modifications, to bevil wheels. The mode of laying down the working drawings of a pair of this kind was explained and figured in p. 120, in relation to the manner of obtaining the form of the teeth; and the proportion of parts may be depicted in connection with the same drawing, by substituting for the primitive cones there given, a section of the pair as represented in the fig. annexed.

Fig. 73.



Here AP is the common pitch line of the pair, determined as before described; and BP, CP are the radii of the wheels. The breadth of the rim of the wheel is marked off from P towards A, and through that point the perpendicular *bb'* is drawn. Another perpendicular to the same line is drawn through the point P, and of course parallel to *bb'*. Upon these perpendiculars to the pitch line AP, the lengths of the teeth are set off, and also the thickness of the rims, as indicated by the lines *ab*, *a'b'*. The distance *a a'* will be the length of the tooth *plus* the bottom clearance; the space inclosed between *a* or *a'* and the pitch line, the length of the flank of the tooth; and from the pitch line to the dotted lines at *a* and *a'* is the measure of the addendum of the teeth. The thicknesses of the rim and arms are in the same relation as for spur wheels; but in this case they are not placed opposite the middle of the rim and boss as before, but entering on one side of the smallest diameter of the wheel, with the feather entire and projecting outwards, so that a cross section of the arm complete would present the appearance ⊤, the horizontal line indicating the face arm, or web, and the vertical piece the feather of the same. The boss or centre of the wheel is built in the manner already described, and of the same relative proportions, but altogether on the side which is furthest from the apex of the ideal cone, of which the wheel is a frustum. The feather of the arm is inserted upon the same side, and is of equal breadth to the face arm. This is commonly sweeped by a double curve, as in the diagram, fig. 73, but often it is simply tapered from the rim till it meets the central boss, which, in well-proportioned wheels, is usually equal to the breadth of the rim *plus* the thickness of the face arm.

The mode of building the rim, which, with the exception of the teeth, forms the only practical difference between wheels of this class and spur wheels, will be understood from the annexed figures, which show the order of position of the segmental rings in plan and section. The segments of the ring marked 4 being cut to the proper breadth, according to the thickness of wood used, they are sprigged to the face-plate, and dressed on their exterior surface, as before explained. But in the operation of dressing, their internal edge is commonly cut away to the diameter at which the next course marked 2 is to be laid. This operation is not indeed essentially necessary, but it affords to the workman a better guide than a mere surface line described with the required radius, and is done without trouble.

Fig. 74.

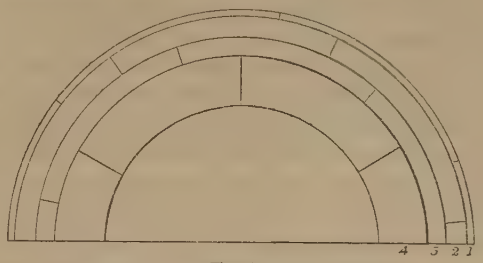
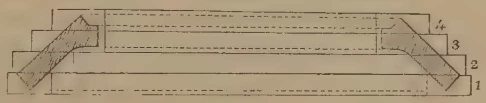


Fig. 75.



The succeeding courses are sprigged and glued on in succession, every succeeding course being increased in diameter according to the angle which the side of the cone forms with the axis—in other words, according as the bevil of the wheel is more or less. Usually the first two courses are *flush* on their inside circumference, as shown in the sectional diagram fig. 75, above.

The rim being thus built, is next to be dressed on its inner circumference to the diameter required. This done, the face-arms are half-checked into the radial part of the ring marked 3, exactly as in the case of the spur wheel pattern. The centre and feathers of the arms are inserted in succession, and before the rim is removed from the face-plate. But as soon as these operations are completed, a centre-plate is fitted to the pattern, and concentric with it; the face-plate is then removed, and the pattern is fitted on the lathe by the centre-plate, for the purpose of having the face of the rim turned. This must necessarily be true to the required bevil, and therefore implies additional care on the part of the workman whose usual method is quite exact. In the first place, he turns down the edge of the course marked 1 to the required diameter; then having fixed for the apex of the imaginary cone a point or edge, he applies a straight edge from time to time during the process of turning, when it is evident that so soon as his straight-edge coincides with the whole breadth of the rim, and at the same time touches the apex and base of the cone, the operation of turning is complete, and not before this coincidence is attained.

An easier method is, however, to turn the greatest and least diameters in succession—with which the drawings furnish him—and afterwards to dress down the face of the rim until a short straight-edge touches the two diameters at the same time, and consequently coincides with the face of the rim throughout its whole breadth. If this be not deemed sufficiently exact, a *mould* may be formed from the drawing, of two slips of wood so joined together as to coincide at the same time with the face-arm and face of the rim; the greater of the two diameters being struck, as in the method first proposed, it is evident that the mould will, on being applied, coincide with the lines on which it was found only when these agree with the drawing. This is perhaps the method most to be preferred, in as far as it is more simple than the first, and more accurate than the second; and it moreover insures the pattern being made exactly according to the drawings, which the other methods do only in respect of the bevil of the rim.

The rim being finished by one or other of the methods described, the next part of the operation is to make the dovetail-grooves which are to receive the dovetails of the teeth. These are cut with the same attention to accuracy of taper as was used to determine the bevil of the rim; for as the teeth ought manifestly to diminish in thickness as they extend towards the apex of the cone, where, if continued to that point, they would vanish, it is consistent with the general character of the work, although not essentially necessary, that the dovetails upon which they are to be fastened take the same tapered form. As in spur-wheel patterns, those dovetails ought to pass through and take hold of each of the several courses of segments of which the rim is built, and thereby act as keys to bind them permanently together.

When the dovetails are fitted and the teeth fastened to them, they are firmly driven into their seats; and the teeth are dressed in the lathe on the ends and points to the required size. The external and internal pitch lines, that is, lines coinciding with the surface of the imaginary cone of which the wheel is a frustum, are drawn upon the ends of the teeth. Upon these lines the centres of the teeth are marked off, next their thickness, and lastly their form is described either by the method of curves, or by pattern teeth shaped according to the mode of development explained in page 121. The dovetails are then driven out, and the body of the teeth accurately dressed to the curves traced on the ends. But this part of the operation ought to be performed in the successive manner directed for spur wheels, and with the same attention and precaution to prevent the *twisting* of the rim, which is almost invariably the result when the teeth are fitted consecutively.

It very seldom happens that two castings are to be made from the same pattern with the same size and form of eye. Every different size of shaft—rather every different diameter of boss upon which the wheel is to be keyed—renders it necessary to change the size of the eye of the wheel; and sometimes the eye is to be bored out and fitted tight with a *feather-key* upon the boss; at other times it is made sufficiently large to admit keys between it and the boss of the shaft; and not unfrequently it is required to be made square to fit square shafts. It becomes therefore necessary to have a ready mode of altering the size and form of the centre according to the size and form of eye wanted. This is provided for by making the original and solid centre smaller than any boss likely to be required for a wheel of the particular pitch. This allows of a temporary centre being built around the original one, of the particular size wanted, and as the eye is always taken out by a *core*, a *print* of corresponding diameter is sprigged on to guide the founder in the size and also in the setting of his core.

It is also necessary often to have the boss and arms more on one side of the wheel than the other, a case which cannot be provided for by making these loose. To modify the pattern for a casting of this kind, the difference is made up temporarily, on the side to which the increase is desired, by pieces simply sprigged to the proper arms and centre of the pattern; the quantity to be removed from the other side is then carefully marked off by chalk lines drawn upon the feathers of the arms and round the centre; and when the pattern is moulded, the parts so chalked off are filled up in the sand. If the modification, however, be of such extent as to alter the position of the face-arms, it then becomes necessary entirely to inclose the arms with thin deal, forming a species of box, of a depth determined on the one side by the position of the arms and centre of the pattern, and on the other by the position which they are to occupy in the casting. The pattern being moulded in this condition, there is left only large open spaces for the arms and centre; and into these the new arms and centre are built by cores formed by core-boxes made to the particular shape and size desired. This is perhaps one of the greatest niceties of green sand moulding.

For wheels of small diameter and pitch, it is now becoming a general practice to use metal patterns, of cast-iron and brass. In works where small machinery is made, and which are commonly provided with wheel-cutting machines, the rim is usually made of sufficient thickness to allow the teeth to be cut upon it, and yet to leave the proper thickness of metal for strength between the bottoms of the spaces, and the *web-plate* which connects the rim with the eye-boss of the wheel. An example of a wheel of this kind is figured at page 69; but sometimes also the same method is adopted with wheels which are *crossed-out*. This method admits of greater accuracy than that described for large wheels; for before proceeding to cut the teeth of the wheel, the casting may be accurately adjusted to a centre in the lathe, and have the strength of its parts at the same time exactly proportioned to the pitch of the teeth. The centre wheel thus admits of being polished, a process which not only improves the appearance, but likewise insures correctness of the work.

At page 70 we briefly described the variety known as mortise-wheels, that is, cast-iron wheels with wooden teeth, or *cogs*, fitted into the rim, and retained in their places by pins or keys as before explained. The mode of constructing the pattern of a mortise-wheel, differs from the ordinary routine for toothed patterns in little more than the omission of the teeth. The rim is however made wider than that of a toothed wheel of the same pitch, by twice the thickness of tooth; and its thickness is double that of the ordinary toothed rim.

The manner of construction is this: The rim is built and dressed, and has the arms and centre fitted into it as already described. The dovetails are also fitted into the corresponding grooves in the rim; but these, instead of having teeth fixed upon them, carry simply thin pieces to serve as core-prints to guide the founder in laying in his cores to make the mortises for receiving the tails or tenons of the teeth.

These prints are made in length equal to the breadth of the teeth, and in breadth equal to their thickness. When the pattern is moulded the prints leave their impressions in the sand, and these are filled by cores of corresponding size and of a depth equal to the thickness of the rim.

The rim of the casting being carefully *faced up*, or dressed in the lathe, is ready to be cogged. The cog-tenons are in the first place fitted tightly into the mortises of the rim; the wheel is then put into the lathe and the points of the teeth and ends of the tenons are dressed off in precisely the same way as the wooden teeth of a wheel-pattern: the pitch lines of the wheel are then run upon them with a point, and the curves of the teeth being described, the cogs are ready to be taken out and finished by hand; or the wheel may be put into the wheel cutting machine, and cut to the form and pitch required.

Wooden cogs, it will be seen from the rules already given for the strength of the teeth of wheels, ought to be thicker than iron teeth in the proportion of the cube of 14 to the cube of 12; the difference is commonly made greater, and perhaps correctly, as the strain falls principally upon the necks of the tenons of the teeth. This allows of the iron teeth of the wheel with which the cogs are to geer being dressed to the exact form and thickness, and polished on their acting surfaces, to prevent injurious abrasion of the wooden cogs. By dressing, the iron teeth are however diminished in strength, and consequently a pair of wheels of the kind here described ought to be made fully more strong than wheels of the ordinary kind.

When wheels are beyond a certain size, it becomes necessary to cast them in segments, which are afterwards united to form the complete wheel. An example of this mode of constructing an internal wheel, has already been given (page 71); but the same is likewise done with external wheels. The pattern of the rim of a wheel of this kind consists of a single segment, of which the required number of castings are made. The segment is provided at the ends with the requisite means of connecting the parts together, and to the arms which are also commonly cast separately, and bolted to the rim and centre. As the rim in this case may be cast of any degree of thickness without risk of injury to the wheel, it is commonly made with recesses to receive the ends of the arms, which are fixed in their places by keys, bolts, or both, according to the mode of construction preferred by the maker. The segments are also sometimes, and advantageously, dovetailed into each other on the inside at the ends, and the arms are fitted also by dovetail-checks and bolts to the middle of their length. This mode of fitting requires great accuracy of workmanship, but when well executed it possesses a degree of neatness which cannot be otherwise attained. The arms are fitted in the same manner to the centre.

Another mode is to cast each segment with an arm attached to the middle of its length, and to fit these upon the centre in the way described. Thus if the wheel have eight arms, the rim will be composed of eight segments, and the centre will be cast with an equal number of recesses to receive the ends of the arms.

When the wheels again are of an intermediate size, too large to be cast in one piece, yet not larger than will admit of their being cast in two parts, they are then cast in halves, each half complete in itself from the circumference to the centre. These halves are then fitted together to form the complete wheel. The manner of fitting is various. The common way is to cast each half with strong flanges throughout the whole diameter, and to bolt these together. Wheels constructed in this way, must of necessity have an *even* number of arms, so that the division may be effected through the middle of an arm running through the diameter of the wheel. The two flanges, that is, the flange of each segment, will form the feather of that arm, but double the thickness of the ordinary feather.

We have seen wheels put together in three pieces in this way very satisfactorily, but the difficulty of fitting is then very considerably increased. It is also not uncommon to cast the segments with strong flanges only at the rim and centre, and to trust to these alone to hold the segments of the wheel together. This greatly reduces the work of fitting, and we do not see that sufficient strength might not thus be attained.

The method described of casting wheels in parts, is partially resorted to when a wheel is to be fixed upon a shaft already fitted up, and which cannot be released from its couplings without considerable difficulty and expense of time. The mode of casting a wheel for this purpose, is to mould the pattern in the usual way; but when the pattern is taken out of the sand a thin plate of iron, rather two plates equal to the radii of the wheel beyond the eye, are inserted into the mould through the feathers of the two opposite arms. When the casting is made, the wheel can readily be separated into two at the parting formed by the plates. Pulleys are also frequently cast in the same way and for the same reason.

With respect to the material of the patterns of wheels, it is only necessary to observe that the common practice of using white deal for the arms and rim, and hardwood for the teeth, and their dovetails, is both economical in point of time and expense. At one time it was thought that nothing else ought to be used than Honduras mahogany, but the common chestnut tree is found to be equally good and greatly less expensive. Beech and plane-tree are the woods ordinarily used for the teeth, and sometimes hiccory and ash for the dovetails; the former takes a fine, smooth surface, and the latter has a degree of toughness, which well adapts it to the particular end in view, that is, to hold together the segments of the rim. In some shops even fir is used for the teeth; but that wood is objectionable for the purpose, not only on account of its softness and more fibrous surface, but likewise on account of its receiving a less accurate outline of the teeth when these are being drawn in. It answers the purpose however very well, when the teeth are afterwards to be dressed either by hand, or in the cutting machine, to the proper shape and size, as when they are intended to work with wooden teeth. But in all cases where the teeth are expected to require no dressing after leaving the sand, it is of importance that those of the pattern be formed with the utmost accuracy, and have a smooth surface. And it may be here observed that unless in the circumstances referred to, it is not desirable, in point of economy of wear, to dress the teeth, at least to the extent of removing the cast surface, which is by far the hardiest and most durable part, and after working some time, it takes likewise a smoother surface than can be given to the softer metal beneath. The surfaces ought however to be cleaned of any imbedded sand, and excrescences which may be found adhering to them when taken out of the sand; and it is also of advantage to *run-up*, or clean the points of the teeth in the lathe when the wheel is small.

The key seats of the eye are usually cut by the slotting machine; they have a taper of 1 in 64, and a breadth equal to the pitch of the teeth.

SECTION VIII.—ON SHAFTING.

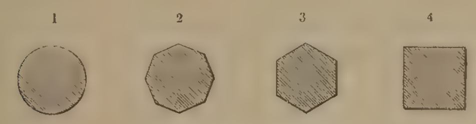
Shafting is a department of millwork which embraces some of the most important considerations within the compass of practical dynamics. In wheel-geering an error in respect of strength is usually attended with less serious consequences, and is more easily rectified than an injudicious distribution and adjustment of the shafts. These are the distributors of life to the whole—the nerves and arteries by which the vitality of motion is transmitted through the system. If on the one hand these organs be too light, and therefore inadequate to perform their functions with steadiness, it will be of little importance that the wheels are accurately made, and proportioned to the power conveyed; the tremor and hobbling gait to which an over-burdened condition of the shaft-geer invariably gives rise, will speedily destroy their adjustment by irregular wearing of the teeth. The bush-brasses at the same time suffer, and the journal becoming loose, the evil is aggravated, till ultimately a sudden failure at some important point sets the whole at rest, and produces an opportunity for renewing a portion of the defective system. Succeeding failures afford new opportunities of displacing piecemeal the worn-out parts; and the process goes on until the whole is finally brought to the necessary degree of strength. But although strength is thus ultimately attained, the geering rarely fails to bear evidence of its piecemeal construction, by want of unity of design in the work, and harmony in the order and distribution of the parts. Although twice made at more than double cost, it still continues unsatisfactory in some respect: it may have sufficient strength, but can neither possess the same degree of elegance and accuracy of adjustment, nor the working convenience which belongs to a system of geering constructed according to a uniform and well-digested scheme, in which the whole is viewed in connection, and not in detached and fragmentary portions. On the other hand, if the shafting be made too heavy, an unnecessary expense is incurred in the construction; and, what is usually of more importance, and in most cases of more serious consequence, a waste of power is occasioned by the unnecessary friction and wear produced by the superfluous weight at the journals and footsteps. Both extremes are thus to be avoided with equal care. It is, therefore, of obvious importance that the millwright should possess a clear knowledge of the principles by which the strength of his shafting may be proportioned to the power which it is required to distribute, and be furnished with safe rules to guide him, without his being reduced to the necessity of recurring constantly to the laborious process of calculating every case *ab initio*, or of determining his sizes by the random mode too often resorted to. It is also of equal moment that he be provided with examples of the various methods of connecting together the shafts of a line, to enable him to compare the different applicable forms, and select with facility that which in point of economy, readiness of construction, and neatness, is best adapted to the particular case in hand.

In no department of practical mechanics have greater improvements been made than in mill-geering, and especially in shafting. Formerly most of the shafts used in mill-work consisted of wood; and possibly that material might still have been retained, had the introduction of the factory system not rendered it of great importance that machinery should be made not only as compact and light as possible, but likewise as durable and as little liable to accident as circumstances would permit. The immense capital embarked in our larger factories renders a stoppage of the works not only a matter of serious importance to the proprietors, but likewise to the multitude of people engaged in them. And as iron is obviously a more enduring material than wood, it is therefore plainly to be preferred both for economy and for lightness and elegance of appearance. The factory system, by affording better and more extended experience to millwrights, has likewise been the means of introducing a higher quality of workmanship—nicer adjustment and better proportion of speeds and strength, than we find in the older systems of geering. One important feature of our modern practice is the higher velocities at which the main shafts are impelled. By this means the geering can be made lighter for the same power, and therefore more durable, since the friction and inertia are diminished, and the impulsion thereby rendered more steady and uniform. Even within the present century the velocities of the main shafts, proceeding from the moving power, did not exceed thirty-five turns a minute, and the speeds of the remoter shafts were seldom as high as fifty turns in the same unit of time. Under these circumstances, the belt pulleys, in order to bring up the speeds to the required intensity at the acting points, were heavy drums, often as much as five feet diameter. By bringing up the velocities of the first shafts to a speed of a hundred and thirty to a hundred and fifty revolutions in a minute, the required momentum is attained with geering proportionately light, and with a diminution of friction directly proportioned to the reduction of weight effected. By these means the power of the prime mover is economised. If that power be derived from steam, a corresponding saving is effected in the maintenance of the engine; and if water power be employed, a proportionally greater quantity of machinery can be impelled by the same waterfall. Hence the principles which ought to guide the engineer in the construction of a system of shaft and wheel geering is, to regulate the connections in such a manner that the inertia of the mass and the friction of the motion may be reduced to a minimum, and to effect this purpose the velocities and strength of the parts must be adjusted to one another, and to the speeds of the machines in which the power is to be expended.

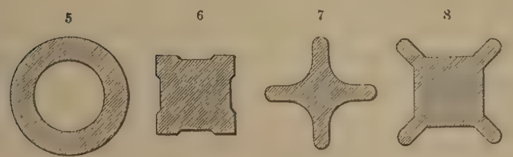
But before entering upon the investigation of the practical bearings of this general principle, it may be necessary to glance briefly at the various kinds of shafting employed by the millwright and engineer. The varieties are not numerous. As respects material, there is only the choice between wood and iron; and the forms are chiefly cylindrical and square, but sometimes octagonal. When large wooden shafts are employed, as they sometimes still are, especially as axles of wooden water-wheels, they are commonly made of one solid log of oak, with gudgeons inserted into their ends, according to a method to be subsequently described. But the scarcity of large oak has not unfrequently led to the substitution of shafts built of fir-planks, and these, when properly made and of sufficient strength, have been found little less durable, and much less expensive.

Iron shafts are of two varieties—forged and cast. Those of large size are commonly of cast-iron, while smaller shafts and *spindles* are formed of malleable iron. In mill-geering the line of demarcation between the two varieties is usually at four inches diameter. The form most common to both is that of a cylindrical solid; but often they are square, sometimes octagonal, and occasionally hexagonal. Cast-iron shafts are also not unfrequently made with ribs upon their peripheries, and then they are called *feathered shafts*—probably from the slight resemblance which they bear to the feathered part of an arrow. Cylindrical shafts of large size are also sometimes cast tubular, and they are then termed *hollow* shafts.

The subjoined figures represent the forms of cross-section most commonly adopted in shafts of the respective materials, iron malleable and cast, and wood.



Malleable iron, cast-iron, wood.



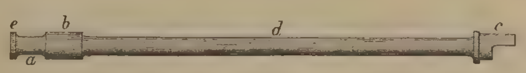
Cast-iron.

In respect of these forms, the cylindrical is in general to be preferred, not that it possesses greater strength as is commonly supposed, for the same weight of materials, but because it is the simplest and the most elegant in appearance when mounted. Feathered shafts, as they are commonly made, although their strength to withstand lateral pressure be augmented by the breadth of the feathers, are very rarely free of tremor from want of substance between the feathers, and unequal distribution of the material around their axes. It is, however, an error to assume with Tredgold and others, that the circle is the only form of section which gives to the axes the property of offering in every direction the same resistance to flexure; for it might readily be proved geometrically that the square section must offer the same resistance to flexure in the direction of its sides and diagonals; and what is true of the square, is equally true of a great number of other figures which may be formed by combining symmetrically the circle and square. The hollow cylindrical form admits of being adopted only in shafts of the very largest class, and for the same weight of material is greatly the strongest form that can be employed, and the best adapted to resist the combined action of torsion and cross strain. For this reason hollow shafts are commonly employed as axles for large water-wheels, where these two forces are brought prominently into action. This form is undoubtedly copied from nature, where we find hollow cylinders opposed to forces which often change their direction, thereby affording the greatest strength with the least weight of material, as in the stalks of the grass tribe of plants, the quills of birds, and the bones of animals. But in artificial works we are unable to pursue the principle with the same advantage; we cannot work with the necessary precision, and find it difficult to make a casting of a tubular form below a certain diameter with the necessary accuracy; and, therefore, in shafts of small diameter it is much greater economy to make them solid.

The sections figured above must be understood to be those of the body parts of the shafts; but the shaft is very seldom uniform throughout its length even when cylindrical, and never when of any other form; for whatever be the figure of the body-part, the journals must always have a circular section, and every shaft must—as will be subsequently seen—have at least one journal or gudgeon on which it revolves. If the shaft be one of a line leading the power to some distant point from the source of motion, it will likewise have its extremities adapted to *couple* with the shafts between which it is placed; and if intended to carry a wheel or belt-pulley, it is commonly provided with a *boss* on which the same is keyed. In very large shafts the boss may be replaced by four ridges projecting from the periphery to serve as key-beds; and when the shaft is of cast-iron and square, as shown by the section marked 4, the part to receive the wheel takes frequently the form of section marked 6.

The following figure illustrates the usual form of a cylindrical iron shaft, on which a wheel is to be keyed at the enlarged part *b*, denominated the *wheel-boss*, and which is to be coupled to another shaft at the part *c*, by what is termed a *half-lap* coupling, to be afterwards described. The journal is marked *a*, and has the same diameter as the body-part of the shaft marked *d*. For bright or polished shafts this is the best, though not the most common practice; and even when the body of the shaft is not turned, the difference of diameter ought not to be more than the least quantity which can be removed by the lathe-tool in forming the journal from the initial body diameter. But we shall have occasion to recur to this part of the subject in speaking of the strength of shafting, and also to the proportions which ought to be

Fig. 76.

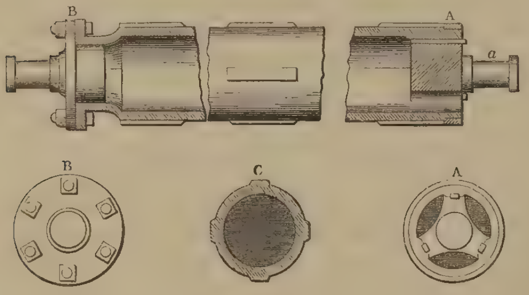


preserved between the body diameter and the wheel and coupling bosses.

In ordinary shafting, the gudgeons are *now* always formed of the same piece as the shaft, and turned. This is not only more convenient than the old practice of inserting gudgeons into them, but is absolutely required by the great accuracy necessary to the high velocities at which the main geering is now driven; for the slightest deviation from the straight line would be immediately destructive of the bearings and toothed geer by the concussive action and unequal momentum produced. In slow motions the evil is serious, but the unequal strain on the bearings is not so immediately disastrous as when the irregularity is several times multiplied; for increasing in intensity as the square of the velocity, the strain is concentrated and developed in a destructive shock upon the full or *lobe* side of the shaft. In ordinary water-wheels, however, the motion at the axis is small, and the axles are generally of large diameter to bear the weight of the wheel. When formed of cast-iron they are usually hollow, with independent gudgeons fitted into their extremities; and when made of wood, the gudgeons must of necessity be inserted.

The annexed cuts show the general form of a hollow cast-iron axle for a water-wheel, with two modes of fitting the gudgeons. The form of gudgeon fitted into the end marked A is peculiarly adapted to axles of this kind on account of its simplicity. The metal of the shaft is here thickened internally to a distance equal to the interior diameter, and three ribs cast in it have key-seats cut in their faces. The gudgeon marked *a* has three arms cast or forged upon it (according as it is made of cast or wrought iron), of the same length as the ribs on the interior of the shaft; and these being dressed in the lathe at the same time that the gudgeon is turned, are correspondingly key-seated. A wrought iron hoop being driven red-hot upon the end of the shaft to prevent it from splitting, the arms of the gudgeon are adjusted in their places and keyed tight, the keys having as little taper as can conveniently be allowed for fitting. They are thus less likely to become loose in their seats.

Figs. 77



Sometimes the gudgeon is provided with four arms, but it is commonly admitted that the mode described is preferable, because with three arms the keys must all bear equally, whereas when four are employed, it is difficult to obtain a uniform tightness of the set.

The form of gudgeon shown at B is more applicable to a vertical than to a horizontal shaft, although employed indifferently with both. The shaft is in this case cast with a projecting flange round its extremity, of the same diameter as a similar flange cast upon the gudgeon. The interior of the end of the shaft is, in the first place, bored out truly cylindrical, and the flange is at the same time faced. The flange of the gudgeon is likewise faced, and a piece cast on the back of it is turned to fit into the turned part of the end of the shaft; the two flanges are then brought together and secured in tight metallic contact by bolts and nuts.

In this form of gudgeon the pivot is sometimes formed of wrought iron or steel, and fitted into the flange and socket, which must of necessity be cast. This allows of the pivot part being replaced when worn out, at less expense than if the whole consisted of one piece.

To render this a good and lasting arrangement, it would be necessary to make the socket of the gudgeon of considerable length, and to bind the two flanges together by a hoop of malleable iron, embracing both their circumferences, and put on red-hot.

When the shafts are of small diameter these contrivances are not usually employed. The part of the gudgeon which enters the shaft is made cylindrical, and the key-seats are recessed in its surface. The cylinder fits loosely into the end of the shaft, and is made tight by the keys, as in the first method described. The shaft, with its gudgeons fitted, is then put into the lathe, and the pivots are turned truly concentric with the body-part, and of the strength desired, the original diameters being usually much greater to allow for any inaccuracy in the fitting. This is also sometimes practised with shafts of the larger class; but more commonly the gudgeon part alone is dressed in the lathe, and the adjustment intrusted to the fitter, who can adapt his keys to obviate any small inequalities due to the casting.

The form of gudgeon most approved of for wooden water-wheel axles is that known as the *cross-tail gudgeon*, which consists of a pivot piece projecting from the centre of a cross, roughly represented by ⊢ in side elevation, and by + on end. The vertical pieces forming the tails are let into mortises cut in the end of the shaft, and the pivot coinciding with the axis of the shaft, the mortises are filled up on each side of the neck by pieces of wood. A strong malleable iron hoop is then wedged tightly on the end of the shaft, in order to fasten and retain the pivot of the gudgeon in its parallel position.

This gudgeon has also been made with only two cross-tails; and in well-finished work the tails, whether two or four, pass within a malleable iron ring, which embraces the shaft at the point where they project to the circumference of the shaft: this ring being made slightly less in diameter when cold, is fitted in its place at a temperature of red heat, and contracting, embraces the tails and shaft tightly. The second ring at the end is made of full size, and fixed by wedges. As an additional security, the arms of the cross are sometimes fixed by screws, which pass through into the wood of the shaft. The arms are in this case made thicker, and do not enter so far into the shaft. The screws by which they are fastened are made of considerable length, and the distance being noted to which they will pass into the timber, mortises are cut at these points large enough to receive nuts suitable to the bolts. The nuts being dropped into their places, opposite to the screwed ends of the bolts, they are screwed up by a spanner, and thereby hold the arms tightly in their place.

By these and similar means the gudgeon can be made for a time quite fast; but when it is considered that the direction of the stress which tends to loosen it is continually changing, and that such stress being exerted on wood, a material which is comparatively very easy and permanently compressed, it is not difficult to perceive that the fastening of gudgeons must have been a matter of much importance to the older millwrights, and that with all their care and ingenuity they seldom succeeded in rendering them firm and lasting. Accordingly several modifications of the schemes described have from time to time been resorted to with indifferent success. The best of these is undoubtedly that of Robert Hughes, described in the Transactions of the Society of Arts, (vol. xxxi. p. 223). "It consists in casting the gudgeon with cross-arms, which fit into notches in an octangular box of cast-iron which has been previously fixed on the end of the shaft." The contrivance is represented by the subjoined cuts, in which "A represents (in section) a portion of the end of a wooden shaft of an octangular form; it is long enough to reach across the pit in which the water-wheel works, and, having a gudgeon at each end, is supported and revolves upon them in proper bearings. BB is the cast-iron box accurately fitted on the end of the shaft, and being wedged tight, prevents the wood from splitting as effectually as any hoops can do. Upon the end of the box is a projecting flange *aa*, with four notches to receive the cross-arms *bb*, *dd* of the gudgeon C. This cross is firmly attached to the box by four screw-bolts, which pass through the flange and the ends of the cross-arms. The figure in section (in which the cross and box are represented separately to show more clearly the mode of putting them together) explains a further precaution which is necessary for strength; and which consists in the inside face of the cross having projections *ee*, which enter the end of the box, and keep the pivot truly in the centre, and prevent any lateral strain on the bolts, which have thus merely to hold the gudgeon fast on to the end of the box. When a gudgeon, fitted according to this method, becomes worn out so as to require replacement, it can be removed by taking off the four nuts, and a new one applied; the gudgeon also being of small dimensions, the cylindrical part admits of being conveniently turned in a lathe, which is a considerable advantage."

Fig. 78.

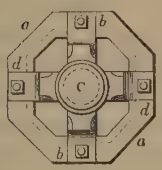
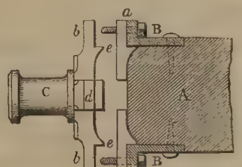


Fig. 79.



This gudgeon involves nevertheless a considerable deal of workmanship, and the real practical advantage which it possesses over the common cross-tail gudgeon is simply, that it does not require the end of the shaft to be impaired by mortising, and affords the means of renewing the gudgeon when the pivot becomes worn out, with more facility and without injury to the shaft.

The stress to which water-wheel gudgeons are subject is generally of a lateral and simple character, particularly when the power is taken off from the circumference of the wheel in the manner before described, (p. 71). The gudgeons have manifestly all the weight of the shaft, wheel, and sometimes the water, to sustain, and ought, therefore, to be made sufficiently strong for that purpose; but they ought obviously to be as little as possible increased in diameter beyond the required strength, and a proper allowance for wear to ensure durability. The strength of a gudgeon, like that of a beam, must evidently be limited by the weight it will bear without permanent derangement of its structure; for the length being small compared with its diameter, the flexure within the point of permanent alteration will be, in all practical cases, insensible. In attempting to determine this limit, the calculated stress should be made to include every kind of force acting on the axles, and the strength made sufficient to enable the gudgeon successfully to resist the whole pressure were it thrown on the extreme point of its bearing.

Now, it may be proved experimentally, that if a cylindrical bar be supported at the two ends in a horizontal position, and loaded at the middle of its length, the diameter which will support a given weight will be expressed by the formula

$$d = \sqrt{\left\{\frac{W \times l}{C}\right\}}$$

in which d is the diameter and l half the length of the cylinder, both expressed in inches, W the load expressed in lbs., and C a divisor depending on the nature of the material. This is exactly the condition of water-wheel gudgeons; and the axle between them being sufficiently strong in relation to the gudgeons to have no sensible flexure, l becomes simply the length of the pivot from the shoulder of the gudgeon, d the diameter, and W the aggregate weight of the axle and wheel. The greatest stress upon the gudgeon takes place when the weight is by any accident thrown upon the extreme point of its bearing; this must be provided for; and we must also make allowance for wear, which, considering the circumstances of the gudgeons being continually exposed to wetness, and gritty matter getting upon their surfaces, may be taken empirically at $\frac{1}{4}$th of the diameter. If then, with this allowance we take the values of C for cast and wrought iron at 1500 and 1600 respectively, our formula becomes

I. For cast-iron $$d\left(1 - \tfrac{1}{4}\right) = \sqrt{\left\{\frac{W \times l}{1500}\right\}}$$

Or $$d = \sqrt{\left\{\frac{W \times l}{633}\right\}}$$

II. For wrought iron $$d\left(1 - \tfrac{1}{4}\right) = \sqrt{\left\{\frac{W \times l}{1600}\right\}}$$

Or $$d = \sqrt{\left\{\frac{W \times l}{675}\right\}}$$

But in practice it is common to make the length of the gudgeon equal to the diameter, that is $d = l$; applying this principle our rules become.

$$\text{I. For cast-iron} \quad d = \frac{1}{25}\sqrt{W}.$$

$$\text{II. For wrought iron } d = \frac{1}{26}\sqrt{W}.$$

These expressions given in words are the following practical rules:

Find the weight in lbs. to be supported by each gudgeon, extract the square root of the number, and that root divided by 25 when the gudgeon is cast-iron, and by 26 when it is wrought iron, will give the diameter and length of the gudgeon from the shoulder to the extremity, expressed in inches.

Example.—The gross weight on the gudgeons of a water-wheel is 86800 lbs.; required their diameter, the gudgeons being cast-iron? The gross weight being 86800 lbs., that upon each gudgeon will be 43400 lbs., and $\sqrt{43400} = 208.32667$, which divided by 25 gives 8.3336 inches for the diameter of the gudgeon. The actual diameter of the gudgeons is in this case 8.5 inches, and they have worked in cast-iron bearings for upwards of twelve years, the bushes being only once and lately renewed during that period.

Mr. Tredgold (Essay on Cast-Iron, Art. 177) has investigated this subject chiefly in relation to a cylinder fixed at one end and loaded at the other, for which he gives this rule:

$$W\,l = .7854\,f\,r.^3$$

When d, the diameter is put for r the radius, of the cylinder, and when f is taken = 15300 lbs. and the length l is assumed in feet, this rule becomes

$$\left(\frac{8\,W\,l}{1000}\right)^{\frac{1}{3}} = d. \qquad \text{Or,} \qquad \frac{1}{5}(W\,l)^{\frac{1}{3}} = d.$$

And applying this to cast-iron gudgeons, making allowance of $\frac{1}{5}$th the diameter for wear, and changing the denomination of l to inches, then results the very simple rule

$$\frac{1}{9}(l\,W) = d.$$

His rule for wrought iron similarly expressed is

$$\frac{1}{9.34}(l\,W)^{\frac{1}{3}} = d.$$

In words—Multiply the stress in lbs. by the length in inches, and the cube root of the product divided by 9 for cast-iron, and by 9.34 for wrought iron, is the diameter of the gudgeon in inches.

For example, the length of a gudgeon is 7 inches, and the stress upon it is 10 tons = 22400 lbs.; then

$$7 \times 22400 = 156800$$

and the cube root of this number is 54 nearly; hence $\frac{54}{9}$ = 6 inches, the diameter required.

By the preceding rule, the diameter of the gudgeon in this case would be 5.987 inches, that is, 6 inches very nearly; and since it requires simply the extraction of the square root of the initial number expressing the stress in lbs., instead of the cube root of seven times that number, it is more easy of application and possesses equal accuracy.

But the author seems not to have been satisfied with the results of this direct rule, for observing that the stress of a gudgeon on its bearing ought to be limited, for otherwise it would wear away very quickly, he proceeds to assume that the stress ought to be confined to a portion of the circumference equal to $\frac{3}{4}$ths of the diameter of the gudgeon; and that the pressure ought to be limited to 1500 lbs. upon a square inch—about the highest pressure we ought to put upon the rubbing surfaces when one of them is of gun-metal. In this case we have

$$l = \frac{4\,W}{3 \times 1500\,d}; \qquad \text{Or,} \qquad l = \frac{W}{1125\,d}.$$

And to allow for a small portion of friction we make

$$l = \frac{W}{1000\,d};$$

If this value of l be introduced into the preceding equation, we have

$$\frac{1}{90}\left(\frac{W^2}{d}\right)^{\frac{1}{3}} = d; \quad \text{or,} \quad W = 854\,d^2; \quad \text{and,} \quad l = .854\,d.$$

This rule gives the size of the gudgeons too small in diameter, and decidedly too short; the length, as before stated, ought not to be less than the diameter, even when the gudgeons turn in gun-metal pillows; and if the pillows be cast-iron, the durability of the bearings, and also of the gudgeons, will be greatly increased by a fractional increase of length. The reason is obvious, for the friction being the same for the same diameter of gudgeon, by increasing the length the surface is thereby proportionally increased, and the rubbing and abrasive action diffused. Instead of the pressure being taken at 1500 lbs. per square inch, it ought not, in circumstances where lubrication is so often defective, to exceed 1200 lbs. with gun-metal pillows, and 1000 lbs. on cast-iron pillows; if these limits be exceeded, abrasion proceeds rapidly whenever the lubrication becomes deficient; and abrasion being once commenced, especially with pillows of the latter kind, it is scarcely possible to stay its progress.

The rule given by Robertson Buchanan in his Essays on Millwork, for the strength of water-wheel gudgeons, affords a ready approximation. It is simply to extract the cube root of the weight in hundred-weights: "the result is *nearly* equal to the diameter in inches of a cast-iron gudgeon sufficiently strong to support such a wheel." The author states that this rule is founded on observation of actual practical cases; but supplies a caution by *nearly*, indicating that the rule is not strictly to be relied upon, and that it would be prudent somewhat to exceed the strength assigned by it.

These remarks, and the rule to which they apply, refer to cast-iron gudgeons, which are commonly preferred in consequence of their being cheaper and more conveniently formed than wrought iron ones. Still there are cases in which wrought iron gudgeons are used, and their diameters may be determined by the rule $d = \frac{1}{26}\sqrt{W}$. We recur to this to guard against the common rule first given by Prof. Robison,* and since frequently repeated,† that the stiffness of wrought and cast-iron being "to each other as 14 to 9 where the permanent change is considered, and as the resistance to this change is as the cubes of the diameters," it is inferred that where the same resistance is required, the diameters in the two cases may be made inversely as the cube roots of these numbers, that is, as 1 to .87; and therefore to find the diameter of a wrought iron gudgeon, knowing that of a cast-iron one of the requisite strength it is only necessary to multiply that diameter by .87. But more recent investigations have shown that the strength of malleable iron referred to cast-iron as unity is only 1.12, and its stiffness 1.3; and it has already been remarked that, in calculating the strength of gudgeons, it is the resistance to permanent alteration which ought to be taken as the proper measure, since they are too short to admit of sensible flexure within the elastic limits; hence the diameter of the malleable iron gudgeon ought to have a diameter = .963 parts of that of a cast-iron one to bear the same stress; for if a be the diameter of the cast-iron gudgeon, and b the diameter of the wrought iron one, then $a^3 \times 1 = b^3 \times 1.12$ when their strengths are equal; consequently

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* Ency. Britanic. Art. Strength of Materials.

† It is repeated by Robertson Buchanan in his Essays on Millwork, and also by Prof. Barlow (without remark) in his generally excellent Treatise on the Manufactures of Great Britain, Ency. Metropolitana.

$$\frac{a^3}{1.12} = b^3. \quad \text{Whence,} \quad 0.963\,a = b$$

Our rules give $0.962\,a = b$, which may be considered identical, yet they are derived from different data.

With respect to the strength of shafts generally, there are different circumstances to be considered in the calculation. Sometimes, as is the case of the water-wheel axle to which attention has been directed, the whole stress may be considered of a lateral kind, tending to break the shaft across; in other cases the stress arises entirely from *torsion* tending to twist the shaft; but it more frequently happens that both these species of stress must be provided against in the same case. Thus all horizontal shafts are liable to the lateral stress at least of their own weight, and that of any wheels, pulleys, and couplings upon them, and likewise to the twisting strain thrown upon them by the power and resistance; but the stress upon a vertical shaft arises entirely from torsion—excepting indeed what may arise from the inaccuracy of the teeth of wheels, and in some few cases where it gives or receives power through the medium of a belt. It frequently happens in examining the conditions under which the shaft is to be placed, even when the stress is manifestly compounded of lateral pressure and torsion, that one of these may be neglected in the calculation of the strength. Thus the effect of lateral pressure is to produce deflection, which must be provided against, and ought not to exceed $\frac{1}{100}$th of an inch in one foot of length; if therefore the dimension requisite to give this degree of transverse stiffness be greater than is required for the twisting power brought on at the circumference, this latter strain may obviously be disregarded; and conversely, when the torsion is great and the lateral pressure insignificant, as in the case of vertical shafts, the effect of the former only requires to be considered. And again, when the shafts are long, and have the power and resistance acting at their extremities, it is not enough that the body part of the shaft be sufficiently stiff to prevent deflection beyond the proper limit; it must have sufficient dimensions to prevent twisting beyond a certain quantity; and in estimating the effect of torsion it is not so much the shaft itself which ought to be considered as the journal which is constantly exposed to wear, and which is thereby rendered more liable to rupture.

In examining successively the amount and effect of these species of stress, and the circumstances under which their intensity is developed separately and in combination, the first part of the inquiry which naturally claims attention is the degree of strength requisite to withstand lateral stress. In this inquiry it is not the ultimate transverse strength, but the stiffness of the shaft which claims attention; in other words, the resistance which the material offers to bending by its own and any superadded pressure tending to deflect it from the straight line. Now, the shaft being square, if

d = the depth or side of the square in inches.
L = the length of the shaft in feet.
W = the weight or pressure upon it in lbs.
δ = the deflection at the middle of the shaft in inches.
M = the modulus of elasticity, taken from the table in page 68, for the particular kind and quality of material to which the calculation refers; we then have

1°. When the shaft is supported at both ends, and the stress is intermediate.

$$d^4 = \frac{432\,L^3\,W}{M\,\delta}; \quad \text{or,} \quad d = \sqrt[4]{\left\{\frac{432\,L^3\,W}{M\,\delta}\right\}}.$$

That is in words—Multiply the weight upon the shaft in lbs. by the cube of the length expressed in feet, and by 432, and divide the product by the value of M taken from the Table, p. 68, multiplied by the assumed amount of deflection δ to be allowed: the fourth root will express in inches the depth of the side of the shaft when the transverse section is square. Or the square root will express the area in square inches of the cross section when the shaft has any other form.

This rule answers for any material of which the weight of the modulus of elasticity M is known. As an example, let it be required to determine the diameter of the oak shaft of a water-wheel; the length is 20 feet and the stress upon it 7 tons = 15680lbs. so that the deflection shall not exceed 0.2 inch.

Supposing the material to be English oak, we find from the Table, p. 68, that M = 1451200; therefore,

$$M\delta = 1451200 \times .2 = 290240.$$

And from the relations of the circle and square, the strength of a square beam is to that of an inscribed cylinder as 1 : 0.589, that is as 1.7 to 1 nearly. Hence, we shall have for the coefficient in this case, 432 × 1.7 = 734.4.

And $L = 20$; therefore, $L^3 = 8000$; consequently,

$$734.4\,L^3\,W = 92123145833.$$

This divided by 290240 gives for quotient 317403, of which the fourth root is 23.736, the diameter required in inches.

This rule, in its general form, is laborious, from the magnitude of the numerical quantities involved, and, in fact, can only be used with facility in connection with a Table of logarithms, to which it is well adapted; but fixing upon particular values of M for the different materials to be employed, and also, fixing the maximum value of δ, it may be greatly simplified. In shafting, as already remarked, the deflection should not exceed $\frac{1}{100}$th of an inch for every foot of length; hence,

I. *For wood*—taking M generally = 1,500,000lbs. and $\delta = \frac{L}{100}$ inches:

Then for square shafts, d being the depth of the side of the square in inches,

$$d^4 = \frac{L^2\ W}{35} \ \ldots\ldots\ldots\ldots \text{(A)}$$

And for round shafts, d being the diameter in inches,

$$d^4 = \frac{L^2\ W}{20} \ \ldots\ldots\ldots\ldots \text{(B)}$$

II. *For cast-iron.*—Taking M = 18,000,000lbs. and $\delta = \frac{L}{100}$ inches:

When the section is square, d being the depth of side,

$$d^4 = \frac{L^2\ W}{412} \ \ldots\ldots\ldots\ldots \text{(C)}$$

When the section is round, d being the diameter,

$$d^4 = \frac{L^2\ W}{240} \ \ldots\ldots\ldots\ldots \text{(D)}$$

III. *For malleable iron*—Taking M = 24,500,000lbs. and $\delta = \frac{L}{100}$ inches as before:

When the section is square and d the side,

$$d^4 = \frac{L^2\ W}{567} \ \ldots\ldots\ldots\ldots \text{(E)}$$

When the section is round and d the diameter,

$$d^4 = \frac{L^2\ W}{334} \ \ldots\ldots\ldots\ldots \text{(F)}$$

2°.—If the stress be uniformly distributed over the length of a square shaft of which l is the length in inches, the fundamental equation is,

$$d^4 = \frac{5\ l^3\ W}{32\ M\ \delta} = \frac{270\ L^3\ W}{M\ \delta};$$

L being the length in feet. From this it appears that a weight uniformly diffused over the length will produce the same deflection of the shaft in the middle as $\frac{5}{8}$ths of that weight applied at that point; for $270 \times \frac{8}{5} = 432$, the coefficient of the fundamental equation under 1°. And if W be the weight of the shaft itself, and it be otherwise unloaded, the deflection produced by its own weight will be,

$$\delta = \frac{5\ l^4}{384\ H\ d^2} = \frac{22.5\ L^4}{H\ d^2}$$

H being the height of the modulus of elasticity expressed in feet, which for malleable iron = 7550000 feet; and for cast-iron = 5750000 feet. By substituting these values in the latter form of the equation we obtain,

I.—For cast-iron shafts when square,

$$\delta = \frac{L^4}{256000\ d^2} \ \ldots\ldots\ldots\ldots \text{(G)}$$

And when the shaft is cylindrical and solid,

$$\delta = \frac{L^4}{150000\ d^2} \ \ldots\ldots\ldots\ldots \text{(H)}$$

II.—In wrought iron shafts when square,

$$\delta = \frac{L^4}{336000\ d^2} \ \ldots\ldots\ldots\ldots \text{(I)}$$

And when the shaft is cylindrical and solid,

$$\delta = \frac{L^4}{198000\ d^2} \ \ldots\ldots\ldots\ldots \text{(J)}$$

As an example of the application of these rules, let it be required to determine the deflection at the middle of the length of a 3 inch malleable iron round shaft of 15 feet length betwixt its bearings?

$$\text{Here } L = 15, \text{ and therefore } L^4 = 50625,$$
$$d = 3, \text{ and therefore } d^2 = 9;$$
$$\text{Hence, } \delta = \frac{L^4}{198000\ d^2} = \frac{50625}{1782000} = .03157 \text{ inch.}$$

The converse of this case frequently occurs; that is, having determined the diameter of the shaft, and assumed a maximum amount of deflection to be allowed, what may be the distance between the bearings?

Let the maximum deflection be $\frac{1}{100}$th of an inch for every foot of length, that is, $\delta = \frac{L}{100}$ inches on the whole length, then from equation,

$$\text{(G)} \ldots\ldots\ldots L = \sqrt[3]{2560\ d^2} \ \ldots\ldots\ldots \text{(K)}$$
$$\text{(H)} \ldots\ldots\ldots L = \sqrt[3]{1500\ d^2} \ \ldots\ldots\ldots \text{(L)}$$
$$\text{(I)} \ldots\ldots\ldots L = \sqrt[3]{3360\ d^2} \ \ldots\ldots\ldots \text{(M)}$$
$$\text{(J)} \ldots\ldots\ldots L = \sqrt[3]{1980\ d^2} \ \ldots\ldots\ldots \text{(N)}$$

Thus, let the diameter $d = 3$ inches, as in the last example, the shaft being of malleable iron and a solid cylinder. Then from equation (N) we have

$$L = \sqrt[3]{1980 \times 9} = 26.12 \text{ feet,}$$

the distance between the bearings, on the condition that the deflection of the shaft by its own weight shall not exceed $\frac{1}{100}$th of an inch for each foot of the length. This being 26 feet in all, the deflection at the middle will be $.26 = \frac{1}{4}$ inch nearly, which is a safe allowance, in ordinary cases, on this length of shaft.

Again, if the torsion be very small, and it be more convenient, and, possibly more economical, to adjust the diameter of the shaft to the limit of deflection than the length of the shaft, this being fixed, then, assuming the value of δ as before, we obtain from equation

$$\text{(K)} \ldots\ldots\ldots d = \sqrt{\left\{\frac{L^3}{2560}\right\}} \ \ldots\ldots\ldots \text{(O)}$$
$$\text{(L)} \ldots\ldots\ldots d = \sqrt{\left\{\frac{L^3}{1500}\right\}} \ \ldots\ldots\ldots \text{(P)}$$
$$\text{(M)} \ldots\ldots\ldots d = \sqrt{\left\{\frac{L^3}{3360}\right\}} \ \ldots\ldots\ldots \text{(Q)}$$
$$\text{(N)} \ldots\ldots\ldots d = \sqrt{\left\{\frac{L^3}{1980}\right\}} \ \ldots\ldots\ldots \text{(R)}$$

Thus, the length between the bearings being 30 feet, what ought to be the size of a cast-iron square shaft in which the deflection shall not exceed $30 \times .01 = .3$ inch at the middle of the length? In this case the rule is equation (O), whence,

$$d = \sqrt{\left\{\frac{30^3}{2560}\right\}} = \sqrt{10.59} = 3.2542 \text{ inches,}$$

the side of the square, the area of the cross section of the shaft being 10.59 square inches.

The distances between the bearings given in these examples are not, however, such as would be adopted in practice; but simply show the limits in reference to the deflection of the shaft by its own weight. The distance between the bearings is determined by other considerations—the prevention of vibration, which would occur, were the bearings placed at distances such as those mentioned, and the number of points at which the power is to be taken off. The spring to which a shaft is always liable in working, from irregularities in the power and resistance, even when its deflection is far within the limit prescribed, renders bearers in general not more than 10 feet apart necessary to prevent vibration; but these are usually of a less substantial character than those in the immediate vicinity of the working points. One cause of vibration consists in this, that the pressure at the acting points, owing to imperfection of workmanship, is rarely tangential to the circumference of the shaft, but operates in part as a lateral force producing lateral deflection, which must be provided for by increasing the number of bearings, and which is not susceptible of being computed, as in no two cases is it the same for the same power and resistance. Inaccuracy in the form of the teeth of the wheels, imperfect adjustment of weight in the revolving parts in the wheels, pulleys, and shafts themselves,—all tend to produce and aggravate the evil, and leave only the two modes of correction open—increase in the number of carrying points, or greatly increased stiffness of the shafts. The former is the more economical mode, and it is accordingly adopted.

III. It is sometimes found unavoidably necessary in a system of geering to *overhang* a wheel or belt pulley, leaving only one end of the shaft supported, while the stress falls upon the other. The equation for the stiffness within the elastic limit is in this case,

$$d^4 = \frac{4\, l^3\, W}{M\, \delta} = \frac{6912\, L^3\, W}{M\, \delta};$$

when L = the length of the shaft from the point of pressure to the bearing, expressed in feet; and M the weight of the modulus in lbs.; d and δ have the same values as before.

Now, in cast-iron, the value of M was before assumed = 18,000,000lbs.; therefore, when the shaft is cylindrical, (the only case which it is necessary to consider) we shall have

$$d^4 = \frac{L^3\, W}{1532\, \delta} = \frac{L^2\, W}{15} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (\mathrm{S})$$

when $\delta = \frac{L}{100}$ inches. This equation determines the least diameter of the journal, which is never greater than that of the shaft.

Again, the value of M for wrought iron is 24,500,000lbs; hence, for round shafts of that material,

$$d^4 = \frac{L^3\, W}{2085\, \delta} = \frac{L^2\, W}{21} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (\mathrm{T})$$

when $\delta = \frac{L}{100}$ inches as before.

As an example of this case, let it be required to determine the diameter of the journal of a malleable iron shaft on which a belt pulley is overhung, at a distance of 2½ feet from the journal? Let the weight of the pulley be 180lbs. and the tension of the belt, when driving, 320lbs.

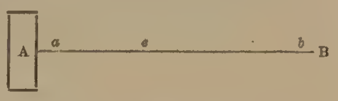
Here L = 2.5 feet, and W = (180 + 320) lbs. = 500lbs.

$$\text{Hence, } d^4 = \frac{(2.5)^2 \times 500}{21} = 148.81,$$

$$\text{And } d = \sqrt{\sqrt{148.81}} = \sqrt{12.21} = 3.49 \text{ inches,}$$

the diameter required.

In this equation it is presumed that no flexure of the shaft takes place beyond the journal, and, to fulfil that condition —which is necessary to save the bearing, and avoid undue friction—the shaft must either be made very strong or have only a short distance between the first and second bearings. Thus, suppose A to be a pulley overhung on the shaft AB, to prevent the shaft from springing between the proper bearings a, b, an intermediate bearing c is placed at a short distance behind that marked a, which prevents upward flexure, and thereby saves the pillows adjoining the stress. It may also be here observed that the distance between the bearings which our rules seem to allow are greater than is commonly practised in modern mill work; steadiness of motion is much desired, and, as friction is not increased by the number of bearings, provided these be properly made and fixed, it is reckoned good economy rather to multiply the bearings than to risk even the small amount of deflection which we have taken as a safe limit.



We have hitherto referred only to two forms of shafts— square and round; and these solid. But the same rules may be considered to apply generally to any other forms of solid section; for knowing the side of a square shaft of a given strength the cross sectional area of any shaft of equivalent strength is thereby approximately determined, however different in form. In a hollow shaft it is in the first place necessary to decide on what aliquot part of the diameter the thickness of metal should be—taking the diameter as 1; then the difference between twice the thickness of metal and 1 will be the aliquot part to be left hollow. For, supposing a solid cylinder to be expanded into a tube of equal length, the quantity of matter remaining the same, the strength will be increased in the ratio of the diameter, and the stiffness in the ratio of the square of the diameter; thus, the strength of the solid shaft being 1, and the diameter of the hollow part being n, the strength of the tube will be $\frac{1 - n^4}{\sqrt{(1 - n^2)^3}}$; and the stiffness being the same as that of a solid shaft of the diameter d, will be expressed by $\frac{d}{\sqrt[4]{1 - n^4}}$. Or more simply, the weight which a solid shaft will sustain with a given amount of deflection multiplied by $(1 - n^4)$ will be the weight which a hollow one of the same diameter will sustain.

As an example, let the thickness of metal be fixed at ⅛th of the diameter; then

$$1 - \frac{2}{6} = \frac{2}{3} = n; \text{ and}$$

$$\sqrt[4]{1 - n^4} = \sqrt[4]{1 - \left(\frac{2}{3}\right)^4} = \sqrt[4]{\frac{65}{81}} = \frac{2.84}{3} = \frac{1}{1.0565}.$$

And if the diameter of the solid shaft be found by the rules answering to the particular case, and that diameter be multiplied by 1.0565, the result will express the diameter which the hollow shaft should have to be, of the same stiffness, the hollow part being $\frac{2}{3}$rds of the whole diameter.

Similarly, when the thickness of metal is $\frac{1}{4}$th of the diameter, the multiplier is 1.032; and for $\frac{3}{20}$ths it is 1.07.

The weight which a hollow shaft will sustain when the thickness of metal is $\frac{1}{5}$th of the diameter, is 0.87 of the weight which a solid cylinder of the same external diameter will sustain; for then $(1 - n^4) = 0.87$; and its stiffness is to that of a square shaft of the same depth, as 1 to 2 nearly.

Mr. Tredgold's rule for the strength of hollow shafts within the elastic limit, the cylinder being supported at the ends, is

$$\sqrt[3]{\left\{\frac{\text{W L}}{500\,(1 - \text{N}^4)}\right\}} = d,$$

and is thus expressed verbally: Fix on some proportion between the diameters; let the external diameter be to the internal as 1 is to N; the number N will always be a decimal, "and ought not to exceed 0.8." Then multiply the length L in feet by the weight W to be supported in lbs. Also, multiply 500 by the difference between 1 and the fourth power of N, and divide the product of the length and the weight by the last product, and the cube root of the quotient will be the exterior diameter d of the shaft in inches. The interior diameter will be the number N multiplied by the exterior diameter, and half the difference of the diameters will be the thickness of metal.

As an example of the application of this rule, let it be required to find the dimensions of a hollow shaft for a water wheel, which, including the weight of the water in the buckets, weighs 44,800lbs.; the whole length of the shaft is 8 feet, from which, deducting 5 feet, the width of the wheel, leaves 3 feet for the length of the bearing: required the exterior and interior diameters of the shaft?

Making N = .7, its fourth power is .2401; and

$$1 - .24 = .76.$$

Therefore, we have by the rule,

$$\frac{44{,}800 \times 3}{500 \times .76} = 354 \text{ in the nearest whole number};$$

and the cube root of 354 is 7, the exterior diameter in inches; and

7 inches × .7 = 4.9 inches the interior diameter.

If the proportion between the exterior and interior diameters be fixed, so that the thickness of metal shall always be $\frac{1}{5}$th of the exterior diameter of the cylinder; then N = .6, and the rule becomes,

$$d = \sqrt[3]{\left\{\frac{\text{W L}}{435}\right\}}$$

which applied to the foregoing example, gives

$$d = \sqrt[3]{\left\{\frac{44{,}800 \times 3}{435}\right\}} = \sqrt[3]{309} = 6.76,$$

the exterior diameter in inches; and the thickness of metal being $\frac{1}{5}$th of this, is $1\frac{3}{8}$ inch nearly.

When the shaft is supported at the ends, and the stress is not in the middle of the length, but at distances p and q from the respective ends, the rule becomes,

$$d = \sqrt[3]{\left\{\frac{4\,\text{W} \times p\,q}{500\,\text{L} \times (1 - \text{N}^4)}\right\}}$$

Thus, let the weight of a wheel and other pressure on the shaft be 36,000lbs. = W; the distance of the point of stress from the bearing at one end being 3 feet = p, and the distance from the other bearing 1.5 foot = q; and let N = .8; required the exterior and interior diameters of the shaft?

The fourth power of .8 is .409; and 1 — N^4 = .591. Therefore, by the rule

$$d^4 = \frac{4 \times 36{,}000 \times 3 \times 1.5}{500 \times 4.5 \times .591} = 485;$$

and the cube root of 485 is 7.86 = d, the exterior diameter in inches; and 7.86 inches × .8 = 6.3 inches, the interior diameter. These rules give the strength of hollow shafts within the elastic limit, but when the deflection is restricted to a given amount, the diameter must be determined from the rule

$$\frac{d = \text{diameter of solid shaft}}{\sqrt[4]{1 - n^4}} = \text{D};$$

the diameter of a hollow shaft of the same stiffness as the shaft of diameter d.

The second species of stress to which shafts are liable is *torsion*—which, by a transposition of the cause and effect, is sometimes defined to be the resistance which the shaft offers to a force applied to twist it round. By torsion, however, we are to understand the effect, not the cause—the arc through which a point on the circumference of the shaft is made to pass by a force acting at that point tangentially. Thus, if a shaft—suppose a cylindrical one—be immoveably fixed in reference to its transverse section, at one end; and a tangential pressure be applied at the other end, at a given distance from the axis, the shaft, under these circumstances, will be subjected to *torsion*, and the forces opposed to the lateral displacement of its opposite parts, and to its rupture, constitute its resistance to torsion. Within the limit of rupture, the pressure and elastic forces called into action must necessarily be in equilibrium, and, observing the torsion produced with a given pressure, we learn the capability of the shaft to withstand stress acting tangentially to its circumference. The shaft being twisted through a certain arc subtending an angle at the centre, which may be represented by θ, if r be the radius, and l the length; also, R the distance from the centre at which the weight W acts; then it may be shown that the *angle of torsion* is expressed by

$$\theta = \left(\frac{2\,\text{R}}{\pi\,\text{G}}\right) \times \frac{\text{W}\,l}{r^4}$$

where G is a coefficient, depending for its amount on the nature of the material, and called the modulus of torsion. From the recent experiments of M. Cauchy, the value of G in relation to the modulus of elasticity M is represented by the simple formula

$$G = \frac{2}{5} M;$$

so that it can readily be determined for any given material from the Table in page 68—observing that for the values of G thus taken, all the dimensions given in the foregoing equation must be taken in inches, and the weight in lbs.

The question of torsion divides itself into two cases as applied to shafts; for, if the shaft be very short the amount of twist will only be limited by the ultimate strength of the material, but in ranges of shafting of considerable length, the angle of torsion must necessarily be confined within certain limits, depending upon the degree of accuracy of motion, requisite in the particular instance. In the first case it is usually the strength of journal which is to be determined, for the journal being of sufficient diameter to resist the force applied to it tangentially, the body part of the shaft will be in no danger of rupture from that cause, since it has at least equal strength, and is not subject to wear; and, moreover, the ultimate strength must obviously, as respects torsion, be independent of the length—provided, indeed, the length be not less than the diameter of the shaft. For shafts of which the lengths are small, and in which the angle of torsion may consequently be neglected, we have the following rules:

Putting W = the stress in lbs. tending to twist the shaft,
" R = the radius of the wheel or pulley, at the circumference of which the power is applied, expressed in feet.
" d = the diameter of the shaft at the weakest part (the journal) expressed in inches.

The rule for a solid cylindrical or polygonal shaft of cast-iron is,

$$d^3 = \frac{W\,R}{125}; \quad \text{whence } d = \tfrac{1}{5}\sqrt[3]{W\,R} \cdots\cdots\cdots (\text{s})$$

which, for similar wrought iron shafts, becomes,

$$d^3 = \frac{W\,R}{140}; \quad \left.\begin{aligned} \text{whence } d &= \sqrt[3]{\frac{W\,R}{140}}; \\ &= .1783\sqrt{W\,R} \end{aligned}\right\}\cdots\cdots(\text{T})$$

If the body part of the shaft have less strength than the journal, and have a square section, the rules become,

$$\text{For cast iron,} \quad d = \sqrt[3]{\left\{\frac{W\,R}{150}\right\}} \cdots\cdots\cdots\cdots\cdots\cdots (\text{U})$$

$$\text{For wrought iron,} \ d = \sqrt[3]{\left\{\frac{W\,R}{168}\right\}} \cdots\cdots\cdots\cdots\cdots\cdots (\text{V})$$

In the same manner it is shown that, for hollow cylindrical shafts of cast-iron and of short lengths the resistance to torsion is expressed by the equation,

$$d^3 = \frac{W\,R}{125\,(1 - n^4)} \cdots\cdots\cdots\cdots\cdots (\text{W})$$

where $n = \frac{d - 2\,t}{d}$, the thickness of metal being expressed by t.

When the cylinder is of malleable iron, and bored hollow, as is the case with the main-shaft of the Great Britain steam ship, the equation becomes,

$$d^3 = \frac{W\,R}{140\,(1 - n^4)} \cdots\cdots\cdots\cdots\cdots (\text{X})$$

To exemplify the application of these rules, let it be required to find the necessary diameter of a water-wheel axle from which the power is taken off at the circumference of a spur wheel of 16 feet diameter placed upon it; the axle to be cast-iron, and capable of conveying the power of 2000 cubic feet of water descending per minute upon the circumference of an overshot wheel of 20 feet diameter from centre to centre of the shrouding, and having a circumferential velocity of 5 feet per second.

In this case, the force to produce torsion of the axle may be considered to act either on the circumference of the water-wheel, or with a radius equal to that of the spur wheel, by which the power is given off; for, the moving force and resistance being equal, the calculation may manifestly be applied to either indifferently.

The head of water reduced to the diameter of the water-wheel will in this case be 20 feet, and its mechanical value at the given velocity will consequently be,

$$\frac{2000 \times 62\tfrac{1}{2}\left(20 - \frac{5^2}{2\,g}\right)}{33{,}000} = 74\tfrac{1}{4} \text{ H.P nearly.}$$

Hence, the pressure to produce torsion of the axle will be,

$$\frac{74\tfrac{1}{4} \times 33{,}000}{5 \times 60} = \frac{2{,}450{,}250}{300} = 8167\tfrac{1}{2} \text{ lbs.}$$

and this weight acts with a leverage of 10 feet taken at the water-wheel; consequently,

$$d^3 = \frac{W\,R}{125} = \frac{8167.5 \times 10}{125} = 653.2.$$

Therefore, by a Table of powers and roots, we find,

$$d = \sqrt[3]{653.2} = 8.676,$$

the diameter of the axle in inches when solid.

If the shaft be intended to be hollow, and to have a thickness of metal t, equal to $\frac{1}{5}$th of the external diameter, then,

$$n = \frac{d - 2\,t}{d} = \frac{3}{5}$$

$$(1 - n^4) = 1 - \frac{81}{645} = \frac{564}{645};$$

$$\text{and } 125\,(1 - n^4) = 125 \times \frac{564}{645} = 109 \text{ nearly;}$$

and equation (W) thus reduced, becomes

$$d^3 = \frac{W\,R}{109} = \frac{81675}{109} = 749;$$

$$\text{therefore } d = \sqrt[3]{749} = 9.082 \text{ inches,}$$

the external diameter of the axle; and the thickness of metal being $\frac{1}{5}\,d = \frac{1}{5} \times 9.082$ inches $= 1.816$ inches.

This case serves to illustrate generally the mode of calculating the strength of shafts to resist a force tending to twist them, when the lengths are such that the angle of torsion may be neglected. It comprehends the two conditions which occur in practice; and when the shaft is hollow adopts a thickness of metal which affords a good proportion. The rules, for convenience, may be thus verbally expressed for cylindrical shafts, the only form necessary to be attended to in practice.

I. *For solid shafts to resist torsion—Equations* (S) *and* (T).—Find the pressure in lbs. acting upon the shaft at the circumference of the wheel or pulley receiving or transmitting the power: multiply that pressure by the radius of the wheel or pulley, that is, by the leverage at which it acts; if the shaft be cast-iron, divide that product by 125, but, if malleable iron, divide it by 140; and extract the cube root of the quotient; this root expresses in inches the diameter of shaft corresponding to the given pressure and leverage.

II. *For hollow cast-iron shafts, the thickness of metal being* $\frac{1}{5}$*th of the diameter—Equation* (W).—Find, as before, the product of the pressure and leverage: divide that product by 109, and take the cube root of the quotient as the required diameter in inches.

For particular cases in which a different thickness of metal is employed, the equation, of which the rule is only another form of expression, must be reduced to find the proper divisor. Thus, supposing the thickness of metal to be fixed at $\frac{1}{7}$th of the diameter, then

$$n = \frac{d - 2\,t}{d} = \frac{5}{7};\ \text{and } n^4 - \frac{625}{2401};$$

$$\text{therefore, } 125\,(1 - n^4) = 125 \times \frac{1776}{2401} = 92.5;$$

which must be taken as the divisor instead of 109 when the thickness of metal is reduced from $\frac{1}{5}$th to $\frac{1}{7}$th of the diameter of the shaft.

In cases where malleable iron tubes are employed on account of lightness, or other property or convenience, the equation (X) may be similarly reduced for the different thicknesses of metal used.

These rules are necessary and sufficient to determine the strength of short shafts to resist twisting, and, consequently, ought to be employed in calculating the strength of journals; but in shafts of great length in comparison with their diameters the angle of torsion becomes an important element in the investigation.

Now, if the extension which the material will bear by twisting without injury when the length is taken as unity be assumed,

$$\text{for cast-iron} = \frac{1}{1200},$$

$$\text{for wrought iron} = \frac{1}{1400},$$

then the value of θ in our fundamental equation, that is, the angle of torsion, will be,

$$\text{for cast-iron } \theta = \frac{2284\ \mathrm{L}}{1000\ d};$$

$$\text{for wrought iron } \theta = \frac{1965\ \mathrm{L}}{1000\ d};$$

in which L is the length of the shaft in feet; d the diameter in inches; and θ the angle of torsion in degrees of a circle.

Thus, if a vertical shaft of cast-iron, 6 inches diameter, and 36 feet in length, be strained to the extent of its elastic force, the angle of torsion will be,

$$\frac{2284 \times 36}{1000 \times 6} = 13.7 \text{ degrees.}$$

This amount of twisting could not, in ordinary cases, be permitted in practice; but it shows the range within which our calculations must be confined; and the limit of absolute safety as respects strength to resist torsion. The rule, moreover, indicates the condition that the angle of torsion is as the length directly, and the diameter inversely; and may, therefore, be adjusted with precision to the degree of accuracy with which the motion of the first mover must be transmitted; for, although in certain cases of irregular moving powers—as wind and horses—a considerable degree of elasticity may be of advantage; yet, in factories where cotton and other textile manufactures are conducted, and in which every irregularity is necessarily guarded against, only a very limited amount of elasticity of the shafting is permitted.

Taking the modulus of elasticity of the two varieties of iron at the same values as before, our fundamental rule becomes, for round shafts of

$$\left.\begin{array}{rl} \text{cast-iron and solid } d^4 = & \dfrac{\mathrm{L\,R\,W}}{53.5\,\theta}; \\ \text{,, and hollow } d^4 = & \dfrac{\mathrm{L\,R\,W}}{53.5\,\theta\,(1 - n^4)} \end{array}\right\} \ldots (\mathrm{Y})$$

$$\left.\begin{array}{rl} \text{wrought iron and solid } d^4 = & \dfrac{\mathrm{L\,R\,W}}{71.3\,\theta}; \\ \text{,, and hollow } d^4 = & \dfrac{\mathrm{L\,R\,W}}{71.3\,\theta\,(1 - n^4)} \end{array}\right\} \ldots (\mathrm{Z})$$

These are the rules of most importance, as cylindrical shafts are now almost uniformly adopted; and, moreover, they apply without much error to the hexagonal and pentagonal forms. For square shafts of s inches on the side, the rules will stand for

$$\left.\begin{array}{rl} \text{cast-iron } s^4 = & \dfrac{\mathrm{L\,R\,W}}{90.5\,\theta} \\ \text{wrought iron } s^4 = & \dfrac{\mathrm{L\,R\,W}}{121\,\theta} \end{array}\right\} \ldots\ldots\ldots\ldots\ldots (\mathrm{AA})$$

The data usually attainable in practice are the power which the shaft is required to transmit, the leverage at which the power acts, and the length of the shaft; it then remains to fix the degree of torsion which may be permitted without injuriously affecting the regularity of the motion: these quantities being settled, the rules expressed by (Y) and (Z) for solid cylindrical shafts are thus applied:

Multiply the power in lbs. by the leverage at which it acts, and by the length of the shaft, both in feet; divide the product obtained by 53.5 or 71.3 times the number of degrees in the angle of torsion allowed, according as the shaft is of cast or wrought iron; and the fourth root of the quotient will be the diameter of the shaft in inches.

Thus, by a line of shafting of 40 feet length, the power transmitted is 500lbs. acting at the circumference of pulleys of 1 foot radius: required the diameter so that the angle of torsion shall not exceed 2 degrees at the extremity, the shafts being wrought iron?

$$\left.\begin{array}{ll} \text{Here } 500 \times 40 \times 1 & = 20000 \\ \text{and } \quad 71.3 \times 2 & = 142.6 \end{array}\right\} \text{then } \frac{20000}{142.6} = 140.25.$$

And the fourth root of 140.25 is 3.44, so that the shafting would be made $3\frac{1}{2}$ inches diameter—which is the actual case.

The rule for hollow shafts differs from that given only in this, that the thickness of metal in relation to the diameter must be assigned as before explained; and for square shafts, the rule differs only in having different coefficients of the angle of torsion.

These rules are sufficient to meet all the cases of calculation which occur in practice relative to the strength of shafts; but the mode of expression often causes considerable previous computation to determine the value of the power transmitted in lbs. as here required. The more common and convenient mode is therefore to estimate the strength and sizes of the shafting by the horse-power transmitted, and the velocity. According to this measure, from what has already been explained in respect of velocity, it will be perceived that the resistance must be estimated as the horse-power *directly*, and as the number of revolutions *inversely*, since, with a given power, the velocity must be greater, as the resistance is less. Thus, the resistance due to 32 horse-power on a journal making 50 revolutions per minute will strain it to the same, and only to the same extent, as the resistance due to 16 horse-power acting on a journal which makes only 25 revolutions in the same unit of time; for $\frac{32}{50} = \frac{16}{25} = .64$. And, in all cases, when the horse-power, divided by the velocity, gives the same quotient, the stress is the same.

If the rule contained in equation (Y) for cast-iron shafts be reduced to the notation corresponding to the dynamical unit of power, it assumes the form,

$$d^3 = \frac{\mathrm{H}}{\mathrm{N}} \times \frac{25.5\ \mathrm{L}}{\theta} \quad \cdots\cdots\cdots\cdots\cdots\cdots \text{(BB)}$$

in which H represents the number of horse-power transmitted by the shaft, and N the number of turns which it makes in a minute; L and θ are respectively the length and angle of torsion expressed in feet and degrees as before. From the rule in equation (Z) for wrought iron shafts we have the form,

$$d^3 = \frac{\mathrm{H}}{\mathrm{N}} \times \frac{20\ \mathrm{L}}{\theta} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots \text{(CC)}$$

These rules will apply to the ordinary kinds of shafting, and where it is necessary to keep the amount of torsion in view; but, in cases where less exactness is required, the simpler rules furnished by Robertson Buchanan may be adopted. These rules are intended to comprehend three classes, as—1st. steam engine fly-wheel shafts; 2nd. shafts in immediate connection with water-wheels; and 3rd. shafts for ordinary internal mill geering. The following are the rules:—

1st. class $d = \sqrt[3]{\left\{\frac{\mathrm{H}}{\mathrm{N}} \times 400\right\}}$

2nd. class $d = \sqrt[3]{\left\{\frac{\mathrm{H}}{\mathrm{N}} \times 200\right\}}$

3rd. class $d = \sqrt[3]{\left\{\frac{\mathrm{H}}{\mathrm{N}} \times 100\right\}}$

These rules are stated to be derived from "observation of shafts in actual use, and of acknowledged good proportions"—and for short shafts, they express pretty nearly the practice of some of our best mill-wrights; but, in the last edition of the Essays on Millwork the rule for cylindrical cast-iron shafts is

$$d = \sqrt[3]{\left\{\frac{\mathrm{H}}{\mathrm{N}} \times 240\right\}}$$

These rules apply to vertical shafts, and may also be employed when the lateral pressure is very small; but, in other cases, it may be necessary to determine the size of the shaft from the sum of the straining forces.

We have already found, that taking the strength to resist torsion within the limits of elasticity, for a solid cylinder the diameter is expressed by

$$d = \tfrac{1}{5} \sqrt[3]{\mathrm{R\ W}}\text{, and therefore, } 125\ d^3 = \mathrm{W},$$

when the twisting stress is collected at the surface of the shaft. But in that case $\mathrm{W} = 740 \times \frac{\mathrm{H}}{v}$, where v is the velocity of the surface in feet per second, and H the greatest number of horse-power that is required to be transmitted through the shaft; consequently, if the velocity of the surface of the shaft be equal to its circumference in feet multiplied by the number of revolutions r, in a second, we shall have

$$v = \frac{11\ r\ d}{42};$$

and equating the stress and strain with this value of v, we have

$$d^3 = \frac{3.767\ \mathrm{H}}{r}.$$

And comparing this with equation (D) it appears that when

$$\sqrt[3]{\left\{\frac{3.767}{r}\right\}} = \sqrt[4]{\left\{\frac{\mathrm{L}^2\ \mathrm{W}}{240}\right\}}$$

the strain from twisting will be equal to that from lateral pressure; and, therefore, if $\frac{1407}{\mathrm{L}^2} \times \left(\frac{\mathrm{H}}{r}\right)^{\frac{4}{3}}$ be less than W, the diameter of the shaft ought to be determined by the rule for lateral stress; if it be greater than W, the diameter must be found by the rule for torsion. This calculation is complex, but, in ordinary cases, when the stress in lbs. multiplied by the square of the length in feet, is less than 3000, the twisting strain will be the greater, and the converse.

In a series of horizontal shafts, it is necessary to make the journal equal to the twisting strain with a proper allowance for wear; and, hence, it is often necessary to adjust the shafts to both strains—in other words, to take the sum of the straining forces, which gives,

$$d^3 = \frac{240\ \mathrm{H}}{r} + \frac{\mathrm{L}^2\ \mathrm{W}}{240\ d} \quad \cdots\cdots\cdots\cdots \text{(DD)}$$

But as both members of this equation contain d, in attempting to reduce it, the result is an equation of the fourth degree, which can only be resolved by a complex operation. To avoid this, it is necessary, in the first place, to assume d in the second member equal to 1, when we shall have

$$d^3 = \frac{240\ \mathrm{H}}{r} + \frac{\mathrm{L}^2\ \mathrm{W}}{240}$$

and which will lead to a result somewhat in excess, except when d is less than 1. To correct this result, the approximate valve of d so found, must be employed as a multiplier of 240, and a new solution effected with it as a known quantity. A value of d more nearly correct will thus be obtained, and if still greater accuracy be desired, a third solution may be effected as before. The approximation may be thus continued to any degree of exactness.

As an example, let the following extreme case be assumed:—A cylindrical cast-iron shaft making 120 revolutions per minute, transmits 20 H. P; the length of the shaft is 21 feet, and the lateral pressure 546lbs. when reduced to the middle of the length; required the diameter necessary to resist both pressures?

Here $r = 120$ and $H = 20$; therefore, $\frac{240\ H}{r} = 40.$

Also, $L = 21$ and $W = 546$; therefore, $\frac{L^2\ W}{240} = 1003.$

$$\text{Hence, } d^3 = \frac{240\ H}{r} + \frac{L^2\ W}{224} = 1043.$$

$$\text{And } d = \sqrt[3]{1043} = 10 \text{ the diameter in excess.}$$

Let now this value of d be employed as in the equation (DD) and

$$d^3 = 40 + \frac{21^2 \times 546}{240 \times 10} = 140;$$

therefore, $d = \sqrt[3]{140} \doteq 5.2$; the second approximation in defect. Assuming this last as the correct value of d, then

$$d^3 = 40 + \frac{21^2 \times 546}{240 \times 5.2} = 193;$$

therefore, $d = \sqrt[3]{193} = 5.78$, the third approximation of the value of d in excess, but which differs from the exact value determined by further approximations by no more than the $\frac{3}{100}$th part of an inch; the proper diameter of the shaft may therefore be taken at $5\frac{3}{4}$ inches.

This case might have been resolved entirely by the rule for lateral pressure, as the force to produce torsion is small compared with the other; but it illustrates the mode of calculation by the compound rule more fully than a case in which the forces are more nearly balanced.

The rule is founded on those for cast-iron to avoid repetition; but it may be applied to malleable iron shafts by multiplying the final value of d by .96 for the diameter; and, according to some experiments, the strengths to resist torsion may be taken as equal within the limits of elasticity; and one good authority, M. Morin, makes the strength of cast and wrought iron shafts in the ratio to each other inversely as 2 to 3 within the elastic limit.

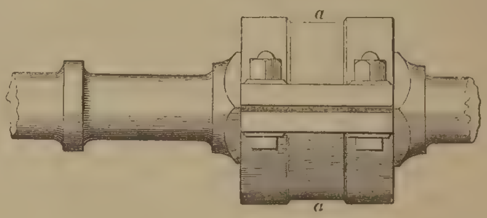
From these considerations of the strength of shafts, we naturally pass to the more technical subject, of the means of connecting several lengths of shaft together longitudinally. These means are denominated

Couplings.—Couplings are of necessity employed in a line of shafting when of greater length than is found practicable to cast or forge in one continued shaft. They are also frequently required in cases where one length of the shaft would suffice, for the purpose of occasionally disconnecting parts of the geering beyond a certain point of the line; and likewise for connecting and disconnecting particular machines. These are the general objects of couplings; and apparently simple in themselves, and for a very simple object, they have exercised much mechanical ingenuity; and couplings, possessing various degrees of merit, have successively been contrived and applied, and, after a time, have fallen into disrepute, and more frequently on technical grounds than from any defect in their principle.

Were the shafts, to which couplings are applied, to continue with respect to each other precisely as they were fitted up at first, the formation of a coupling would be an easy mechanical problem; but, seeing that shafts deviate considerably from their first condition, owing to the wear of the material, sinking of foundations, and such like causes, it becomes necessary to keep these circumstances in view in the construction and application of couplings.

The most simple species of connection is the square coupling, represented in fig. 80. In this the ends of the shafts to be connected are made square, and are embraced by a square *coupling-box*, *a*, the internal surface of which is fitted exactly to the squares of the shafts. The box is divided into two parts, which close together diagonally upon the shaft, and are provided with flanges at their junction, by means of which they are bolted together, and to secure the shafts, so that one cannot turn without driving the other. The coupling is sometimes made quite plain; embracing the shafts like the one now described; and, when occasion requires it, the box may be slipped back on one shaft, to leave it clear of the other, thereby admitting of the motion being discontinued, or of one or both being removed for repairs or alterations.

Fig. 80.

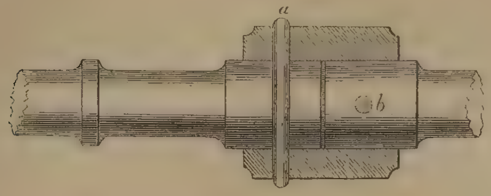


An objection to this arrangement of a rigid square coupling at once suggests itself, when we reflect that, although the motion would go on all very smoothly so long as the shafts remained mathematically true to each other, when the one of them wears down in its bearings faster than the other, or when the wearing is in different directions, it must follow, that in some part of the revolution the shaft is lifted off its bearings, where there are two bearings, one on each side, and unsteady motion is produced, together with farther straining and wearing of the couplings. This objection, it is true, applies principally to the shafts of heavy mill-work; but it is only for such purposes almost exclusively that this form of coupling is employed; in small machinery it is only occasionally employed.

A variety of the square coupling consists in having fitting-strips, or projections along the corners of the square parts of the shafts. The square form of the shaft is in this case virtually preserved, and there is the advantage of the coupling being fitted with greater facility, while the strain is concentrated upon the corners of the shaft.

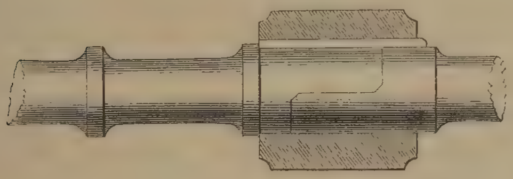
The round coupling is, in some respects, open to the same objections as the square coupling. It will be understood from fig. 81, which represents it in section. In this the ends of the shafts are made cylindrical and meet together, with flat ends, under a round coupling fitted truly upon them, and secured to them by pins *a* and *b* at right angles to one another. This joint may be made with greater precision and much less expensively than the other, as the parts may all be accurately turned and fitted together. It is clear, however, that, as the strain is concentrated on the pins and holes, these parts must wear out soon; at the same time, it is easy to renew the pins, though, obviously, they cannot be made to fit the holes so accurately as they did at first. This coupling is not often used for heavy shafting, but with parallel keys instead of the cross pins, it is not unfrequently adopted for light shafting under the name of the *thimble-coupling*.

Fig. 81.



The half-lap joint, represented in fig. 82, affords a neat and compact method of coupling. The shafts, being cylindrical at the ends, are formed with semi-cylindrical extremities, and fitted together so that the tongue of the one fits into the recess of the other; the cylindrical form is thereby completed, and the joint is covered with a thimble or ring secured to it by a key.

Fig. 82.



This coupling is much employed for all kinds of shafting, large or small, where only one bearing is used. It is exceedingly strong and durable; this will be evident on considering that the key extends through the coupling, and affords a broad holding surface. The coupled part is rendered, in fact, the strongest part of the shaft. The cross section of the half-lap being made equal in area to that of the body of the shaft, it is equally stiff, and, having a greater diameter, it has more strength. The laps are, moreover, tightly embraced by the coupling box, which is driven hard upon them, allowing no *play* at the joint.

This species of coupling is formed partly by turning and partly by planing. The coupling bosses are first turned with great accuracy; they are next planed on their flat surfaces to an exact semi-cylindrical form. The coupling box being turned externally, and bored out to a gauge, is fitted upon them by adjustment of the flat faces of the laps with the file; the key-seats are then cut, and the key fitted, when the operation is complete. The operations are thus attended with considerable expense; but the accuracy and durability, and likewise the elegance of the coupling, fully compensate for the labour bestowed on its original construction. This is acknowledged by its almost universal adoption in the better quality of modern millwright work.

For convenience of disengaging the connection, the boss of the shaft nearest the prime mover is provided with a thin ruff, to which the coupling box is driven up, and which prevents it from passing off in that direction; the adjacent boss of the other shaft is left plain, and from this end the key is driven. To disengage the coupling, the key is driven back by a *drift* passing over the ruff, above which the end of the key rises in thickness; the box is then driven back in the same direction, that is, towards the left in the figure, and relieves the laps. The shaft thus disengaged, when lifted out of contact with the tongue of the other, will cease to be impelled, and, consequently, the motion of all the geering behind that point will be discontinued.

The coupling shown in figs. 83 and 84 is constructed of two cast-iron plates, *aa* and *bb*, keyed one on the end of each of the shafts. The plate *a* is formed with two segmental openings in it, which will be better understood from fig. 84, at *c*, *c*; these openings are intended to receive corresponding projections on the face of the plate *b*, as represented in the section; and thus the shafts become engaged. The rim of the plate *a* overlaps and embraces the circumference of the other plate, and thus they are preserved exactly concentric.

Fig. 83.

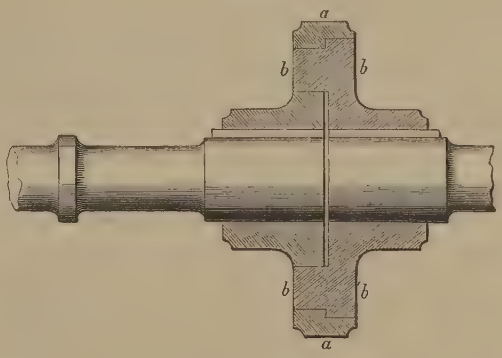
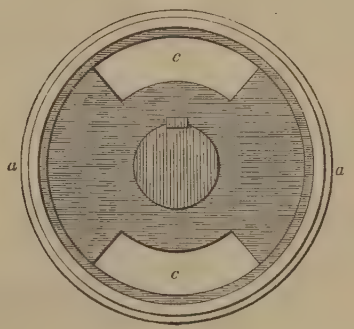


Fig. 84.

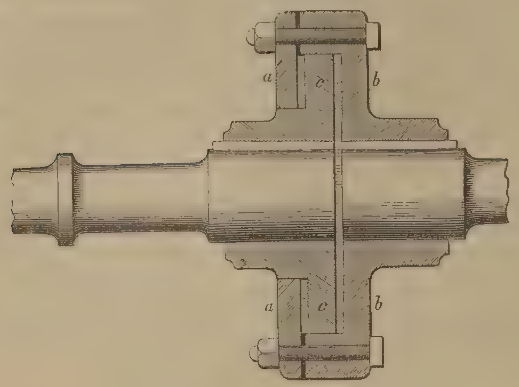


This coupling, applicable to shafts with two bearings, one on each side of the coupling, is simple and durable. It is easily adjusted, and may be disconnected without difficulty;

but care must be taken that both holding surfaces be in contact, otherwise a deal of twisting and straining of the bearings must be the consequence. The larger the radius at which the acting surfaces operate, the more readily may they be adjusted, and the more accurately they are likely to work.

Fig. 85 is another combination of discs suitable for couplings with only one bearing. The disc *b* is keyed on one shaft, and is recessed on the face, to receive the smaller disc, *c;* this disc is sunk flush with the face of the other, and is screwed tightly up to it by means of the ring *a*, which is bolted to the disc *b*, and secures that marked *c*. Between the three plates, *a*, *b*, and *c*, annular pieces of leather are interposed, which brings them all to a proper bearing.

Fig. 85.



This combination, termed a friction coupling, is useful for preventing breakage of the connections in case of a sudden stoppage or reversal of the motion. It is plain that the holding power of the coupling depends simply upon the tightness with which the discs are screwed together, and the consequent frictional force of the surfaces of leather and metal.

Another form of permanent coupling is that known as Hooke's Universal Joint, from the name of the inventor, Dr. Robert Hooke. The object of this coupling is to unite shafts which are inclined to each other in the line of direction, and which do not, therefore, admit of being rigidly connected, as in ordinary cases. This coupling is very commonly employed in light machinery, as in steeple clocks, for taking off the index-motion, and is then usually constructed by forming an arc on the two extremities which it is intended to connect, and to form the joint by a central cross $b \overset{a}{\underset{a}{+}} b$; the extremities of the arc on the end of one shaft being jointed to the arms *a a*, and the extremities of the arc on the other shaft on the arms *b b*, at right angles to the former. But this simple mode of construction is not adequate to the purposes for which the coupling is required in a line of shaft-geering; in this case, although the principle is not in any way changed, the construction is much more substantial. Figs. 86 and 87 represent a form of it adapted to heavy strains. A is a strong disc keyed on the end of each shaft carrying a pair of bearings for the reception of the gudgeons formed on the extremities of the cross. Fig. 87 is a face view of one of the discs, showing the cross in its place, with its alternate journals disengaged.

Fig. 86.

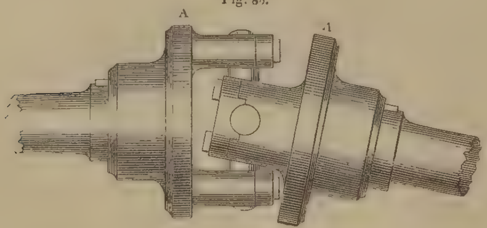
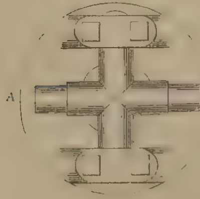


Fig 87

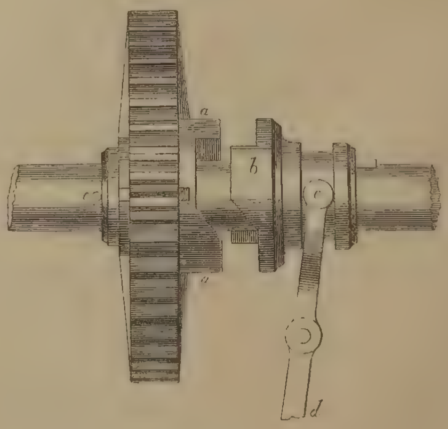


In this illustration the shafts are shown at nearly the limit of the angle to which the single joint ought to be applied. This angle ought not to exceed 15 degrees; when a higher angle is introduced, the rotatory motion becomes very sensibly irregular, and the friction is greatly increased. This defect may be obviated by using a double joint; but a far more simple mode is that depicted in page 73, fig. 16. This is a form of flexible coupling which is perfectly regular in its action, works with little friction, and is, in general, much more simple in its construction. Both of these forms of coupling require, of course, two bearings supporting the two connected extremities of the shafts.

Besides these more permanent forms of couplings there are other contrivances adopted, when the object to be attained is the engagement and disengagement of certain parts of the machinery or geering during the course of operations.

With the same view of admitting of this disengagement of the connection, in cases of sudden stoppage or reversal, the coupling, fig. 88, is sometimes employed. In this instance, the shaft is supposed to be continuous, and the coupling may be termed a *disengaging coupling; a* and *b* are the two parts of the coupling, formed on the acting faces into alternate projections and recesses, such that they corre-

Fig. 88.



spond with, and exactly fit into each other when in geer. The part *a* is, in this example, cast on a spur wheel, from which the motion of the shaft is supposed to be taken off. Both of the parts *a* and *b* are, to a certain extent, loose on the shaft; the former being capable of moving round on it, though deprived of longitudinal motion by washers and pins marked *e*, and the latter being free to slide on the shaft, though prevented from turning on it by a sunk key, which slides in a slot inside the *clutch* or sliding piece *b*. The mechanism is put into geer by means of the handle *d*, which terminates in a *fork* with cylindrical extremities *c*; and it is obvious, that by the contact of the flat faces of *a* and *b*, the latter will immediately carry with it the other part at the same speed as the shaft. Supposing, now, that the motion of the wheel *a* is suddenly accelerated, the oblique faces of the couplings immediately fall out of contact, and slide free of each other, leaving the couplings clear, and the shaft free to continue in motion.

In the old form of this contrivance, known as the sliding bayonet clutch, the part *b*, instead of the tooth-like projections on the face, had two or more prongs which laid hold of corresponding snugs cast on the face of the part *a*—which, moreover, was usually a broad belt pulley introduced with a view to modify the shock on the geering on throwing the clutch into action. In an older form still the pulley was made to slide end-long on the shaft. A form analogous to this was known as the "lock pulley," a few specimens of which still remain in the older factories. Instead of the end-long motion common to the other modes, the parts were "locked" together by a bolt fixed upon the side of the pulley, and which, when shifted towards the axis, engaged with an arm of a cross, of which the part *b*, in the preceding figure, is the modern representative. The bolt was wrought by means of a key and stop, the turning of the key throwing back the bolt, and thereby unlocking and disengaging the pulley.

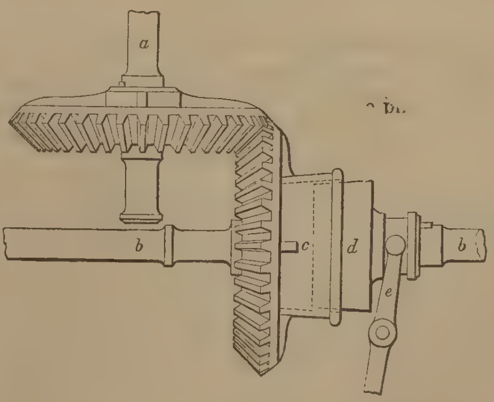
The form of coupling represented by fig. 88 is particularly applicable when the impelling power is derived from two sources—a circumstance which frequently occurs in localities affording water power to some extent, and yet not in sufficient abundance for the demands of the work. The deficiency is usually supplied by a steam-engine; and the two powers are concentrated in the main line of shafting by a coupling of the kind depicted. In cases of this kind, the speed of the shafting being fixed, and the supply of water inconstant, the power of the water-wheel ought to be thrown upon the wheel *aa*, and that of the engine upon the shaft at another point. By this arrangement, the speed of the line can be exactly regulated by working the engine to a greater or less power, according to the supply of water. The proper speed of the water-wheel will likewise be maintained, which in of importance in economizing the water power.

The same form of coupling is also used occasionally for engaging and disengaging portions of the machinery. But for this purpose the object is to obtain a mode of connection by which the motion may be commenced without shock; for, in consequence of the inertia of all material things—that is, the tendency which every portion of matter has, when at rest, to remain at rest, and when in motion, to continue to move—the parts of the mechanism, when acted upon too suddenly by a moving power, are liable to fracture and disarrangement. It is a law in mechanics that when a body is struck by another in motion some time elapses before it is diffused from the point struck through the other parts; consequently, if the parts receiving the blow have not sufficient elasticity and cohesive force to absorb the whole momentum of the striking body till the motion be transmitted to the centre of rotation, fracture of the body struck must necessarily ensue. Hence, in a system of mechanism, any parts intended to be acted upon suddenly by others in full motion ought not only to be strong, but they ought to be capable of yielding on the first impulse of the impelling force with as little resistance as possible, and gradually bring the whole weight into motion. The common mode of driving by belts and pulleys accomplishes this object very satisfactorily. In this the elasticity of the belt comes into action; and being thrown upon the pulley by the strap-guide or fork, it continues to slip till, by the friction between the sliding surfaces, the belt gradually brings the quiescent pulley into full motion. This mode of connection is unexceptionable when the power to be transferred is not great; but its application to large machinery is attended with inconvenience.

When the belt connection is employed, the provision of the fast and loose pulley is the most simple and effective form yet devised for the purpose. It consists simply of two pulleys in juxtaposition on the same axis—the one fast, and the other loose, so that the belt which transmits the motion may be shifted at pleasure upon one pulley or the other, by that means putting in or out of motion the axis upon which they are placed. The driving pulley—that is the pulley on the shaft from which the motion is derived—ought to be equal in breadth to the two on the second shaft; and these last ought to have their rims slightly rounded or swelled in the middle to prevent the belt slipping off—which it is apt to do when the rims of the pulleys are flat. This curious property is of great practical importance, and has obviated all those clumsy appendages formerly required to keep the belt from being thrown.

This mode of driving is not, however, always convenient, and, accordingly, many attempts have been made to accomplish the same with wheels. Perhaps the best of these is

Fig. 89.

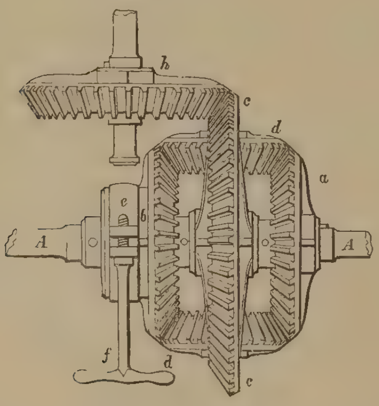


the method of friction-cones, represented in fig. 89. The two parts of this coupling, *c* and *d*, are arranged on the shaft *b*, in the same manner as we have described under the preceding figure. *a* is a shaft driven by means of bevil-geer off the main shaft *b*, its motion being derived from the latter shaft through the coupling. *c* is an interior cone cast upon the back of the bevil wheel; *d* is an exterior cone having the same taper as the cone *c*, such that by means of the handle *e* it may be moved into contact with the interior cone. The surfaces being supposed to be well fitted to each other, the cone *d* will, by its friction, drive the cone *c*, and thereby, also, the upright shaft. When either of the shafts *a* or *b* is accidentally stopped, the cones immediately fall out of geer, and the connection is broken. They are held in geer by means of a screw, or more commonly, and perhaps better, by a weight.

This coupling works very well, if properly adjusted at first; but requires some nicety in communicating the exact degree of taper to the cones; for, if on the one hand, the taper be too small, they are liable to adhere into each other too firmly, and, on the other hand, if the taper be too great, they do not possess sufficient frictional force to keep them in contact.

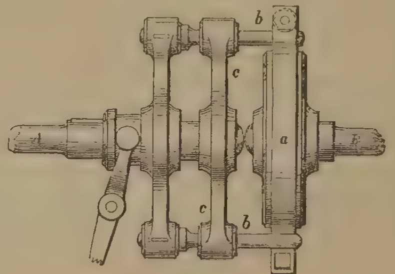
Another mode of accomplishing the same purpose in small machinery, by means of an epicyclic train, is represented by the annexed figure. In this the shaft AA is continuous, and supposed to be that through which the motive power is transmitted. The wheel *a* is fast, but those marked *b* and *c* run loose on this shaft. The two pinions *dd* have their bearings *in* the wheel *cc*, and geer with the two opposite bevil wheels *a* and *b*. (One of these pinions only is requisite to complete the motion; the second being introduced merely to maintain the equipoise of the system.) If now motion be given to the shaft AA, it is clear that the wheel *b*, which is loose, will be made to revolve in the contrary direction to the wheel *a* which is fixed, by means of the carriers *dd*; but no motion of the wheel *c*, *if slightly opposed*, will ensue; and so long as this last remains at rest, the wheels *a* and *b* will have the same angular velocity in opposite directions. But if the motion of the wheel *b* be opposed by means of a friction gland *e*, which can be tightened by means of the T-screw marked *f*, to any degree required, the teeth of that wheel will serve as fulcra to the carrier pinions *dd*, which, becoming levers of the second kind, with the resistance at their axes. will thus carry round the wheel *c* with half the velocity of the prime mover *a*; and *c* geering with the wheel *h*, on the main spindle of the machine to be impelled, will transfer to it the motion which itself receives. We have supposed the wheel *b* to be held absolutely still; but it is obvious that it may be brought gradually to rest by means of the friction gland; and as the wheel *c* can attain motion only as the motion of the wheel *b* is reduced, and can attain its full speed only when *b* is brought to rest, it is clear that the wheel *h*, and, consequently, the machine, may be brought into action without the slightest degree of shock; and, moreover, may be driven at any velocity less than the maximum, that may be desired. This last may, indeed, be looked upon as the principal advantage of the mechanism, and was that which the writer had mainly in view in contriving it. His object was to obtain a speed which would vary according to the resistance, from a line of shafts of which the speed was constant, and which he could also adjust at pleasure within very wide limits.

Fig. 90.



Another mode of obviating shock in starting machinery, which has been long in use, is represented by fig. 91. On the shaft B is fixed a drum or pulley which is embraced by a friction-band *a* as tightly as may be found necessary; this band is provided with projecting ears with which the prongs *bb* of a slip-clutch on the driving shaft A can be shifted into contact. This clutch can be shifted end-long on its shaft A, but is connected to it by a sunk key, so that being thrown into geer with the ears of the friction band, the shaft being in motion, the band slips round on its pulley until the friction becomes equal to the resistance, and the pulley gradually attains the motion of the clutch. The arms and sockets *cc* which are keyed firmly on the shaft A, are used to steady the prongs and to remove the strain from the shifting part.

Fig. 91.

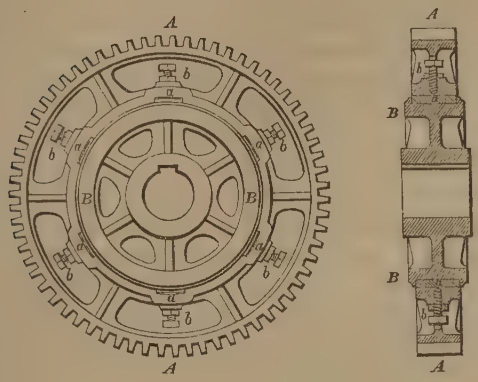


This contrivance is susceptible of very extensive application even in large machinery, and may be variously modified. If the tension of the band be properly adjusted, all shock will be obviated, and the pulley, and, consequently, its shaft, brought into motion in the same easy way as by a belt.

The principle of this arrangement is frequently, and very advantageously applied directly to the wheels themselves through which the power is conveyed. Such wheels are termed Friction Wheels, and their form and mode of operation will be understood by reference to the annexed figure.

The wheel A, which is usually the largest one of the train to which this arrangement is to be applied, is made with an eye as large as can conveniently be used. This eye is accurately bored out and made of such a diameter as to fit the centre or friction wheel B, which is turned truly cylindrical, and is furnished with projecting flanges as shown in the section. In the eye of the wheel, between each arm, small recesses are made, and into these are fitted the pieces of brass *aa*. The pinching screws *bbb* bear upon the back of these pieces, and regulate the pressure to which it may be deemed necessary to subject the wheel. It is obvious that, should the resistance at the teeth of the wheel at any time exceed this pressure, the central part B will revolve and leave the wheel itself stationary, thus saving the machinery from fracture. An example of the application of this arrangement, which will be familiar to most readers, is seen in dredging machinery, which, having frequently occasion to encounter such intractable objects as rocks and trunks of large trees, would be constantly liable to destruction but for this simple and very efficient contrivance.

Fig. 92.



There is still another mode of accomplishing the same end, which is worthy of notice, on account of its simplicity. The contrivance to which we refer consists in transferring the motion of one pulley to a second in contact with it, by means of friction alone. The oldest and best known arrangement on this principle is to make the contact at the circumferences, which are usually composed of end-wood, so that the pulleys may be considered as wheels with indefinitely small teeth. The objection to this arrangement is the amount of friction produced at the journals by the pressure necessary at the point of contact of the circumferences to prevent slipping. To obviate this objection it has been attempted, and with partial success, to make the pulleys concentric, and to bring the two contiguous sides into contact. This modification has been used in hoisting tackle; but while it avoids the friction on the journals, it brings into action an end-long pressure which must be counteracted, and which, consequently, brings into operation a corresponding frictional force. Still there are many cases in which it might be employed with advantage.

Another important feature in a system of mill-geering is the mode of supporting the shafting—the form, arrangement, and general adaptation of the plumber-blocks and steps and their supporting apparatus. These are manifold in their proportions and construction—depending, as they do, in a great measure, on the taste of the engineer and the circumstances in which they are employed. Even the length of journal in relation to its diameter is far from being a fixed quantity; and as the journals determine the length of the *pillows* or *brasses* employed, these last are equally indefinite in their proportions. The range of variation is gradually becoming less as the principle of friction is becoming better known; but still we find journals made equal in length and diameter; and others having a length equal to their circumference. These proportions, except for particular cases, are extremes to be avoided. When the journal, and, consequently, the pillow, is short, the wearing process goes on quickly; the brasses are speedily worn out; and, moreover, while in operation, the waste of oil by heating is unduly increased. Short journals have, in general, been adopted on the erroneous assumption that the friction is thereby diminished—for even now engineers are to be found so little conversant with the principles of their profession as to believe that friction is augmented in proportion to the increase of the rubbing surfaces; and the converse. The short journals are also adopted from a false economy; they are, doubtless, cheaper in first cost, but ultimately entail an expense for maintenance far exceeding the saving primarily effected; and, besides, they invariably defeat the ostensible apology, commonly, though ignorantly made, for their reduced length—namely, economy of power; for, the friction, instead of being less, as supposed, is actually increased by the greater abrasion to which the brasses are subjected. Too great length, on the other hand, is accompanied with two evils; the first cost is unnecessarily increased; and if the shafts be liable to deflection from lateral stress, the friction at the bearings, and, consequently, the abrasion of the pillows, are unduly augmented by the amount of deviation from the right line of the shaft, which takes place within the length of the bearing. The brasses are frequently, from this cause, worn irregularly above and below; and when the plumber-block is narrow, they ultimately lose shape; the ends droop, and, as they can, after a time, offer little support to the journal, the same evils occur as if they had been made originally too short.

Without attempting to establish a general rule for the length to be given to bearings, under all circumstances, it may be observed, that the pressure upon the pillows or steps ought in no case to exceed 1,000lbs. on the square inch of rubbing surface. Within this limit the wear of the brasses is moderate; but, when a less intense pressure can conveniently be attained by increasing the amount of surface, without a countervailing evil, it ought to be adopted, even to half the pressure stated. To calculate every particular case which occurs in practice, and to adapt the patterns to it, would, however, be attended with an amount of labour and expense which it is found necessary to avoid; and, accordingly, the mode commonly adopted is to fix upon two extreme cases, and to proportion the intermediate lengths tc these. Thus, taking a journal of 2 inches diameter, and another 12 inches; and suppose 4 inches to be taken as a proper length for the former and 19 inches for the latter; then the lengths for intermediate diameters may be assigned by arithmetical progression. Thus, between 2 inches and 12 inches there are 40 *quarters*, and between 4 inches and 19 inches there are 120 *eighths;* consequently, for every increment of $\frac{1}{4}$ inch of diameter, we have a corresponding increment of $\frac{3}{8}$ inch of length of journal.

The material commonly employed for pillows is a composition of copper and tin in the proportion of 12 to 1. This alloy is much harder than common brass, and is supposed to work with cast and wrought iron with less friction, and to be more durable than most other compounds. But other cheaper substances are occasionally used, and have, in many instances, been found at least equally durable. Even cast-iron pillows, especially when cast in iron moulds, and when the rubbing surfaces are made large in proportion to the pressure upon them, are not inferior in any respect to the bush-metal commonly employed—the friction is, indeed, rather less than greater, and being harder, they are still more lasting. But when the pressure is so great as to produce abrasion of the surfaces when left occasionally without a proper supply of oil, the wearing process, thus commenced, goes on increasing, and the surfaces never afterwards acquire the necessary polish. In this the bush-metal has an advantage, for although abrasion has proceeded a certain way, when the lubrication is again perfect, the pillows take a new polish as perfect as before, and the abrasion ceases. Wooden pillows are also sometimes employed with advantage. Box-tree and lignum vitæ were long in use, but those woods, besides being much more expensive, are far inferior for the purpose to home-grown beech. We have known journals which, with ordinary metal pillows, were ever liable to heat, run for many years on beech-wood without manifesting any tendency of that kind. This is, indeed, what might be expected on observing the low coefficient of friction between surfaces of iron and beech-wood when lubricated. Holly has, likewise, been recommended for the same purpose.

Pure tin is, perhaps, of all other substances, that which produces the least friction with iron; but its softness has prevented it being very extensively employed. When the pressure upon the rubbing surfaces exceeds very moderate limits, the tin yields to it, and becomes extended in the direction of the length of the pillow. A mode of obviating this difficulty has recently been patented by Mr. Babbitt of New York, U. S., America; it consists in placing the soft metal in a species of casing of cast-iron, of the common form of the half pillow, but with ledges round its concave surface, which thus presents a recess of a depth corresponding to the size of the pillow—from $\frac{1}{4}$th inch to an inch in very large diameters. This recess is filled with the soft metal which is retained by the ledging of the casing and prevented from yielding and becoming extended by the pressure on its surface.

This simple contrivance has deservedly met with the approval of many of the more intelligent engineers in this country and America—especially those concerned in railway machinery, to which any improvement of this nature is of the utmost importance. The material used by Mr. Babbitt is not pure tin, but a soft alloy, in which lead predominates, and which, besides answering all the purposes of tin, has the advantage of being greatly cheaper.

A patent for a composition-metal has lately been obtained by Mr. Fenton; it is harder than Mr. Babbitt's metal, equally fusible, and still cheaper, and has been found very efficient as bush-metal in railway engines. It is simply a compound of tin and zinc with a little copper to harden it; and a very excellent composition of the sort may be made of equal parts of the two former metals and a sixteenth by weight of antimony to give hardness. However, the old gun-metal pillows will continue to be used in mill-work until some mill owner, more economical and better informed than his neighbours, shall set the example of employing a less costly material; and we know of nothing better than cast-iron for ordinary purposes, if the condition of sufficient surface be attended to.

Forms of plumber-blocks and pillows, very commonly adopted for heavy shafts, are shown by figs. 1, 2 and 3 of Plate I. In this the pillows are made octagonal to prevent their turning round with the shaft, and are cast with flanges on their ends to prevent longitudinal displacement. This form has, however, been departed from by some of our best millwrights. Instead of casting the brasses with flanges and plane faces, they are cast of a cylindrical form, and with two snugs at the middle of their length, which, entering corresponding recesses in the plumber-block, retain them securely in their places when the cover is fixed. By this arrangement they are more easily fitted, contain less useless metal, and have a lighter appearance. The plumber-block itself, also, admits of being bored out to the size of the pillows; whereas, with the octagonal form the faces required to be planed to the required angles—a much more expensive process.

The plumber-block consists of a sole and cover which, when practicable, are independently bolted together, the heads of the cover-bolts being counter-sunk in the sole. The cover is made to check accurately within the cheeks of the block, the surfaces being planed true to each other, so that the stress upon the bolts is reduced to a minimum. The sole has two projecting ends by which to bolt it in its place, as shown in fig. 5 and elsewhere in Plates I. and II. The figure specified is an example of a plumber-block with what is denominated a *false* or *shell* cover—a form often adopted for economy when the pressure is entirely down. In this case there is only one brass, the cover-part being cast with side projections resembling the ends of a brass, and with projections on its upper surface resembling the nuts and ends of bolts of the regular form represented by fig. 1. This cover is not fitted, but is simply laid loosely in its place, and serves merely to preserve the journal from sand and similar injurious matters.

The form of pedestal represented under the footstep-bridge by fig. 15 of Plate II. is one of the best examples, combining strength, elegance, and simplicity of make. It is in every way appropriate for its purpose. The base and cover fit together with a degree of symmetry and character which are not found in the examples above referred to; and the amount of fitting is much reduced. The base and cap being checked together by planing, they are fixed in their relative position by the cover-bolts and bored out to the extreme diameter of the brasses. But instead of a uniform concave surface, the interior of the pillow-seat is formed with a circular recess of small depth occupying about half its length, and leaving only about a fourth of the breadth of the concave surface on each side of this recess to be bored. The object of this is to avoid the necessity of turning the exterior of the brasses for an equal extent at the middle of their length, and thus to allow of their being provided with snugs as already noticed, to secure them from turning round in their places with the journal. The brasses being turned to fit their position in the pedestal, they are bored out concentrical with the outer circle; and the points of the cap bolts and the nuts being dressed, the plumber-block is ready to be bolted in its place.

The foundation plate, upon which the sole of the pedestal rests, is usually provided with *snugs*, set so much apart as to allow of a wooden *key* being driven between them and the ends of the pedestal-sole. The bolt-holes in the sole and foundation are also made oblong, to allow of a small amount of adjustment in setting the pedestal; and, to obtain this latitude of adjustment laterally, and also in the direction of the axis of the brasses, the holes are lengthened in the sole and foundation in the corresponding directions. Adjustment is attained vertically by allowing for a given thickness of wood to be placed under the sole of the pedestal, and which can be increased or diminished at pleasure to obviate any small inaccuracy of workmanship.

In ordinary mill-work, pedestals are usually provided with some form of cast-iron foundation plate upon which they are fixed. This is exemplified by figs. 5, 11, and 14 of Plate I. and by figs. 10 and 15 of Plate II.; and it frequently happens that much ingenuity is necessary to *compose* what is technically denominated a *wall-box* or *fixing* for the proper reception of the number of pedestals required by a confluence of shafting to a point. Fig. 11 affords an example of one of the most common and simplest forms of wall-box, being intended to carry only one pedestal for a shaft which may be supposed either to pass through the wall into which the frame is inserted at that point, or to terminate there. If intended to carry the end of a shaft, it is usually made in the form of a rectangular box with the bottom *into* the wall, and with a shelve at a convenient height to receive the sole of the pedestal, the breadth of which determines the depth of the box. But when the shaft passes completely through the wall, the box is simply a rectangular frame, corresponding in depth with the thickness of the wall into which it is inserted and with the cross-shelve, for the purpose before stated. This makes a neat, convenient, and substantial fixing, and one which is in constant requisition.

Fig. 10 gives an example of a pedestal wall-plate, and which serves the same purpose as the wall-box above described, when the shaft terminates at the wall and does not pass through it. This species of fixing is simply a bracket bolted to the wall by three bolts which pass completely through and are secured on the outside by *wall-washers*, *bonnets*, or *stars*. These are merely plates of cast-iron, sometimes with radial arms to increase the amount of bearing surface, with holes through their centres, to receive the ends of the bolts, which being screwed, are retained by nuts passed over them in the usual manner. Commonly the bolts are entered from the outside, which is generally advisable for convenience and safety. The wall-washers serve the purpose of increasing the amount of surface of the wall acted upon, and thereby allow the bolts to be screwed up more tightly; and the more friable the material of the wall is the larger these ought manifestly to be made. For brick walls a very common proportion between the size of bolt and bonnet, estimated by diameter, is *one* to *six*. For dressed stone-walls, the proportion may be somewhat less; but for rubble work the stellar form ought to be preferred.

The bracket is provided with a projecting sole upon which the pedestal rests, and which is supported at the middle of its length by a rib cast upon it. The sole has a *snug* at each end, between which the pedestal is keyed, and holes through it to receive the pedestal-bolts.

Fig. 12 represents another form of wall-bracket. This form is employed for carrying a shaft *along* a wall, and projects sufficiently to allow space for any wheels and pulleys which may be upon the shaft to work clear of the wall. The bearing is in this case formed in the bracket itself, and the cover is held in its place by two bolts secured in the cheeks of the block by cotters.

As this species of bracket does not allow of any adjustment at the bearing, and as it would be difficult to adjust, if fitted directly upon the wall, it is usually provided with a *wall* or *ground-plate*, as shown in the drawing. This plate is, in the first place, secured to the wall by wall-bolts and washers, in the manner above described; and the bracket is then adjusted and fixed upon it by bolts and keys, in the same manner that the simple pedestal is secured to its seat. But sometimes, and judiciously, its bolts also pass entirely through the wall, and serve to maintain the stability of both foundation and bracket.

Figs. 6 to 9 are forms of pendant brackets intended to carry lines of shafting along the ceiling of a building. Fig. 6 is the simplest form, and is often used to carry the weight of the shaft at points intermediate to those at which the power is taken off, but sometimes also it is the only species of bracket employed when the shafting is light. It is easily fitted, and has very commonly only an under pillow for the shaft, and no cover; but when the shaft has an upward pressure—which it must at any point where motion is transferred to or from it by toothed geer, whose line of contact is in the horizontal plane,—a species of block-brass is fitted into it and retained by a cotter, as shown in the figure. The ends of this block are checked, and the back of it has a groove to receive the edge of the cotter; it is thus effectually prevented from moving on end, and may be forced downwards upon the shaft by advancing the cotter, which is made of a tapering form for that purpose.

This form of gallows has, however, no other recommendation than cheapness. It is inelegant in the last degree, and deficient in stability, is easily twisted out of position in consequence of the oblique action of the weight upon it; and, moreover, the hook into which the brasses are fitted being usually narrow—another feature of economy always associated with it—the brasses soon become loose and allow a hobbling motion of the shaft, not more unpleasant to witness than it is injurious to all the connections of the line, and to the accurate transfer of the power at the working points.

Fig. 7 represents a gallows of a more substantial character, but still inelegant in appearance, and, in some respects, inconvenient. The weight being equally and directly borne by the two supporting points, it possesses all the advantages of strength; it is—at least *may be*—provided with a cover and cover-brass, which are necessary when the bracket is placed in the vicinity of a pair of wheels which geer together. This form of bracket is very commonly used in flax spinning factories, but has the manifest inconvenience that the shaft cannot be taken out of its place except by moving it endlong, which cannot always be done without much trouble arising from the operation interfering with other geering. This inconvenience might, however, be obviated by making the arms of the bracket independent of each other, and connecting them at the lower extremities by the block which forms the under half of the bearing; this might be made to fall into tapering dovetails formed on the inside of the arms and connect them together as securely as if the whole were solid metal. It is admitted that the common form requires less work in the fitting shop; but the convenience which would be obtained by the additional expense of workmanship recommended would more than repay itself on the first extensive alteration of the geering.

Figs. 8 and 9 are two views of a form of pendant bracket, which combines to a considerable extent the two requisites of strength and simplicity; and, moreover, is immeasurably superior to the two forms described in point of elegance and character. The pillar and expanded base of this bracket are cast hollow, as indicated by the dotted lines in fig. 8. The cover is fitted to the pillar by planed faces which check together, leaving space to receive a bevil-edged key which holds the cover in its place when the stress is entirely downwards; but when the bracket is intended to be placed in the vicinity of two geering wheels, the cover and projection on which the under pillow rests are provided with snugs through which a *nib-bolt*, as it is technically called, passes and assists the key in retaining the cover in its place.

This form of pendant is also convenient to the millwright as a pattern. In the two forms above described, any difference of length is attended with extensive alteration of the pattern, but in this, the stalk of the pattern is made at least as long as there is any probability of its ever being required; a series of ferrules are then turned to fit upon the stalk, and of the requisite outside diameter; the necessary number of these are put on to make up the length *less* the length of the octagonal part which forms the bearing; this part is formed of cast-iron, and has also a hole through it to receive the stalk, and when put on and fixed in the proper position the bracket is ready to be moulded, any part of the stalk which may be projecting through being *cut off* in the sand. The process of moulding is likewise simple, and no core-box is required, except when the bracket has a double bearing, which is always the case when it is intended for a point at which a bevil pair meet.

The only objection to this form of bracket is the oblique direction in which the strain falls upon the holding bolts, and which might be obviated by making the cover of a piece with the stem, and checking the under bearing to it by a strong dovetail and single holding-bolt, the office of which would simply be to prevent end-long motion. The stress might thus be brought directly in the line of the axis of the pillar, thereby obviating all oblique strain. This arrangement would further allow of the shaft being in many cases more easily dismounted; for although in taking down a shaft thus suspended, it requires to be lifted only through half its diameter, even that is in many instances not easily accomplished without being provided for originally by allowing sufficient space between the *grounds* and the peripheries of the pulleys which may be mounted on the shaft.

Pendant brackets such as those here described, are sometimes cast with vertical soles, prepared to bolt to beams; but more frequently the soles are horizontal; and in that case are bolted to *grounds* of planking fixed between two adjoining beams by wood-screws. When the brackets can be attached to beams, that mode of fixing is, however, to be preferred; and in that case the sole ought to be formed with a projecting rib to bear against the under side of the beam, and relieve the bolts as much as possible of strain.

The kinds of bearings described are all designed to support shafts placed horizontally; but in almost every system of geering there is one large vertical shaft from which the minor horizontal shafts derive their motion. The bottom extremity of the vertical shaft is supported on a species of bearing technically named a foot-step, of which fig. 15 of Plate II. represents a specimen. In this case, the motion is usually transferred by a pair of bevil wheels, as represented by fig. 2 of Plate VI., and more completely still by the arrangement shown in Plate VIII. In this last the first shaft is marked H; on this is the spur pinion G which geers with the wheel F on the parallel shaft K. On the other extremity of this second shaft is the bevil wheel C, which geers with a similar wheel on the vertical shaft; and through this last wheel motion is communicated to the equal wheel B and the geering on the right. The arrangement represented by fig. 2 of Plate VI. is strictly analogous—the horizontal shaft being the driver and the vertical shaft the carrier of the motion to the superior parts of the factory.

In these arrangements the foot-step of the vertical shaft is carried upon an arch, technically a *foot-bridge*, of which fig. 15 of Plate II. gives a front view, and Plate VIII. a side view: fig. 2 of Plate VI. shows a section of the same through the foot-step box. This bridge rests on a foundation-plate of cast-iron, which in turn is fixed upon a stone foundation more or less substantial, according to the weight and stress which are to be resisted. The pedestal of the horizontal shaft ought to be supported on the same foundation sole; for, if supported on a separate sole, a fault which is sometimes committed by unreflecting engineers, the foundation bolts, and possibly the foundation itself, will in all probability be soon destroyed by the twisting action of the wheels at the fixed points. The foot-step box occupies the summit of the arch; three sides of it are cast upon the bridge, but the fourth side, which forms the *door*, is a separate casting, attached by four bolts which pass right through from back to front of the box, hollow *rolls* being cast on the two opposite sides for that purpose. In fig. 15 of Plate II.

the door is supposed to be removed to show the interior arrangement; fig. 2 of Plate VI. gives a transverse section through the door, showing its thickness; and Plate VIII. shows the door in its place and fixed by its bolts. This door is fitted with great exactness, and the whole interior of the box—at least a sufficient bearing surface of it at the angles—is chipped and filed true: a foot-plate as in fig. 2 of Plate VI. and sometimes a species of hollow block, and sometimes both, as in fig. 15 of Plate II., is fitted into the bottom of the box. This plate or pillow is first planed, and in fitting, its superior surface is set truly square with the sides of the box. The surface of the plate is frequently steeled by case-hardening when the end of the shaft is intended to be rested immediately upon it, as shown in the drawings referred to; but more commonly it is simply a plate of cast-iron on which the bottom of the cup-formed brass is supported. In these drawings the brass is supposed to be without bottom, and to embrace the cylindrical end of the shaft, and support it laterally. This arrangement has its advantages, in so far as the foot plate can easily be replaced when worn out; and the brass may be formed in halves, which will allow it also to be replaced with little difficulty. But more commonly the brass has the form of a deep cup, interiorly; while, exteriorly, it is made in size exactly to fit the box. On one side—that from which the wheel acts—a recess is left in the interior to receive and retain oil in considerable quantity, so that the lubrication may be abundant and prevent heating. With every attention to lubrication it is, however, very difficult, when the weight upon the step is great, to prevent its becoming hot, evaporating the oil, and destroying the rubbing surfaces. Various modes of remedying the evil have been attempted; but the best expedient hitherto tried is to enlarge the surface as much as convenient. Formerly, the end of the shaft was greatly reduced to form the foot—often, indeed, a pin was inserted and made to serve the functions of a foot; but at present, the foot is formed on the same principle as any other bearing—namely, with as much bearing surface as the diameter will allow. But even this is often found too little, and the next resource is to continue the wheel boss and give the foot a conical form, thereby increasing the supporting surface as much as circumstances will admit.

We have also seen attempts to line the brass step with ribs of steel to increase its durability; but without success. A step made on Babbitt's principle, and lined with soft metal, would, in our opinion, offer a better chance of success than any scheme which has hitherto been attempted. But the experiment remains to be made on a proper scale, and with sufficiently good workmanship, to test the principle thoroughly before its being adopted very extensively in mill-work.

Among the more ingenious of the older contrivances for rendering foot-bearings durable was that of attaching a hardened steel plate to the extremity of the foot by a square *tang*, which was driven hard into a corresponding hole in the foot. On the under side of this disc a groove was cut across, the better to distribute the oil which found its way to the bottom of the step by vertical grooves cut in it.

This was found to work very well when there was little lateral stress upon the shaft; but ultimately gave place to an arrangement which is sometimes adopted in modern work, and which consists in inserting a steel pivot into the lower end of the shaft. When this pivot is bound in its place by a malleable iron ring put on hot, the arrangement is one of the best which can be adopted; but the original mode of simply boring a hole in the end of the shaft and inserting the pivot, without any further lateral support, left the foot very weak and inadequate to sustain any considerable amount of lateral stress.

Another scheme, also, sometimes put in practice in modern work, when the shaft is very large, is to make the foot turn on conical rollers, on the same principle as horizontal shafts are sometimes made to turn on parallel friction rollers to diminish the amount of friction, by converting a rubbing into a rolling motion, with only the sliding action of the small axes which retain the rollers in their places. The cones, however, require very accurate workmanship, and therefore become expensive, and are, moreover, very liable to get out of order; they are, in consequence, very seldom employed. A modification of the conical rollers, consisting of a single ball placed under the foot of the shaft, although very rarely tried in mill-work, seems to offer advantages more than equivalent to the additional workmanship which it would entail, by dividing the motion, in some measure, between the shaft and itself, and leaving an open space to receive oil, which would tend materially to keep the foot cool. This arrangement is almost universally adopted in vertical sugar-mills, and is found to answer, although in process of time the ball becomes flattened, and requires to be replaced.

The scheme of the late Mr. Bramah, of supporting the pivot entirely on water, the pressure of which was to be maintained by a small forcing pump and stop-cock, would, if all tear and wear could be avoided, be perfect; but, unfortunately, it holds out little prospect of being successfully applied to the end in question, although there appears little doubt that it will ultimately be found both practicable and efficient as applied to railway turn-tables in which the bearing surface is large, and the only problem to be resolved is the diminution of friction.

Water has been successfully applied to keep the common foot-step cool, by simply perforating the foot-box and running a stream of water constantly through it. An example of this may be seen at the Deanston mills.

SIZES AND PROPORTIONS OF BOLTS.

The bolts employed in securing together the pedestals and parts of the fixings of shaft-geering constitute another important element in mill-work. Bolts may be generally described as pins having a *head* formed on one end, and the other end screwed externally to enter an internally screwed ring technically named a *nut*. When the bolt is passed through holes placed oppositely in two pieces, a thin ring or *washer* being first placed on the screwed end of the bolt, if the nut be then put on and screwed tightly down upon the washer the two pieces will be held firmly together between this last and the head, the *body* of the bolt, or part between, serving as the medium through which the force is maintained by one surface upon that opposite.

Bolts are denominated according to their diameters of the body part,

½ in., ⅝ in., ¾ in., ⅞ in., 1 in., 1¼ in., &c., bolts;

the particular diameter being determined in any case by the supposed tension to which the bolt will be exposed. It is rarely that this force can be estimated even approximately; but assuming it to be known the tension ought not to exceed 1½ ton on the square inch of section. Therefore, if we designate by T the tension in tons, that is, the effort tending to separate the head and nut by tearing the body part of the bolt asunder, and the diameter in inches by D, we have the simple formula,

$$D = \tfrac{3}{4}\sqrt{T}.$$

Thus, supposing the known tension to be nine tons, then,

$$D = \tfrac{3}{4}\sqrt{9} = 2\tfrac{1}{4} \text{ inches},$$

the diameter of the bolt necessary to resist that pressure.

The number of consecutive threads in an inch, that is, the pitch of the screw, is regulated by the diameter of the bolt. For small bolts of about ½ inch the pitch is usually about a sixth of the diameter, but relatively decreases inversely with the increase of diameter.

The following Table is that of the dies and taps of an extensive millwright and engineering establishment in Glasgow, and it does not differ sensibly from the numbers adopted in some other works.

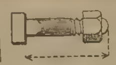
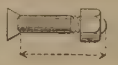
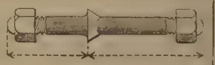
Diameter of screw in inches.	Threads in an inch.	Diameter of screw in inches.	Threads in an inch.
½	12	1¾	5
⅝	11	1⅞	4½
¾	10	2	4½
⅞	8¾	2⅛	4¼
1	8	2¼	4
1⅛	7½	2⅜	4
1¼	7	2½	4
1⅜	6½	2¾	3½
1½	6	3	3½
1⅝	5½	3¼	3¼

The depth of the thread is also a matter of importance. Previous to the recent improvements in the forms of the cutting dies and taps, the thread was only partially formed —was commonly, indeed, only raised by abrasion rather than by cutting the metal. The section of thread most approved of for strength and easy motion of the nut is an isoceles triangle of 55 degrees of inclination of the sides, with the top and bottom of threads slightly rounded off.

Bolts of larger diameter than those named in the preceding Table are usually chased in the screw-cutting lathe, and in that case the thread is commonly made of a rectangular section; but such bolts are very rarely required in millwork, and when they do happen to be employed, it is under such circumstances as remove them from the ordinary class of bolts and the proportions therein recognized.

The length of the body of the bolt must, of course, depend upon circumstances; but a certain proportion is observed between the diameter and the sizes of head and nut. For common bolts the head is usually square, and the nut six paned, that is, has six facets on its periphery. Sometimes, also, the head has the same form, and occasionally it is made round, and for particular purposes it is conical, but oftener pyramidal. When of this shape, it is intended to be *flush* or *even* with the surface of the piece into which the bolt is inserted, and it is then said to be *counter-sunk.* The cover-bolts of pedestals are of this form, being necessarily flush with the sole of the pedestal. The thickness of the head when of the usual form is from two-thirds to three-fourths of the diameter of the bolt; and the diameter of its circumscribed circle is double of the primary diameter. The thickness of the nut is usually made sufficient to contain from eight to ten threads, the smaller having proportionally the greater depth. The part of the body toward the head, equal in length to a third of the whole, is usually made square, and the remaining part of two-thirds is round; and of this a portion equal at least to double the thickness of the nut is screwed. Sometimes, however, as in pedestal cover-bolts, the whole length of the body is cylindrical, the bolt being prevented from turning round when the nut is being screwed up by the pyramidal counter-sunk head.

Although the length of bolts generally can only be assigned for particular cases, those adapted for particular sizes of pedestals may be given. These, it is true, will vary with the particular form of pedestal used; but the sizes given in the following Table will be found very generally applicable for those of good proportions.

Diam. of journal.	Diam. of bolts.	 Holding.	 Cover.	 Double-ended.	
inches.	inches.	inches.	inches.	inches.	inches.
1½	½	3¼	4¼	2¼	5
1¾	⅝	3¼	4¼	2¼	5
2	⅝	3¾	5	2½	6
2¼	¾	4	5½	2¾	6½
2½	¾	4	5¾	2¾	6¾
2¾	¾	4¼	6¼	3	7¼
3	⅞	4½	6¾	3	7¾
3¼	⅞	4⅝	7½	3⅛	8¼
3½	⅞	4¾	8	3¼	8¾
3¾	1	4¾	8½	3½	9
4	1	5	9	3¾	9½
4¼	1⅛	5½	9⅝	4	10¼
4½	1¼	6	10¼	4¼	10¾
4¾	1¼	6⅛	10½	4½	11¼
5	1⅜	6¼	11	4¾	11¾
5¼	1⅜	6½	11½	4¾	12¼
5½	1⅜	6⅝	12	5	12½
5¾	1½	6¾	12¾	5¼	13
6	1½	7	13¼	5¼	13¾
6¼	1⅝	7¼	13¾	5½	14½
6½	1⅝	7½	14	5¾	15
6¾	1¾	8¼	14½	6	16
7	1⅞	8¾	15	6¼	16½
7¼	2	8⅞	15¾	6½	17
7½	2	9	16¼	6¾	17¼
7¾	2⅛	9¼	16¾	7	18
8	2⅛	9½	17¼	7¼	18¾
8¼	2¼	9¾	18½	7¾	19½
8½	2½	10	19¾	8	20¼
9	2⅝	10½	20	8¼	21
9½	2¾	11	20¾	8½	22

Bolts intended to be fixed in stone and called *batt-bolts* are made of larger diameter and barbed, so that being sunk into the stone and a quantity of melted lead poured into the hole, the barbs prevent the bolt from being withdrawn by their hold on the lead. Bolts, again, which are intended to pass through a mass of foundation building, and which can only be entered from above, have usually no head, but are made fast beneath by a cotter passed through the end with a strong washer between it and the stone to give the necessary surface bearing. The heads of bolts intended for wood are commonly segments of a sphere; and the bolt is not often provided with a nut, the holding being in the wood itself. But sometimes, when the wood is thin, the screw is passed sufficiently far through to take a broad washer and nut above it, an arrangement which gives strength in proportion to the acting surface.

Below we give a Table of the proportions of bolts and nuts employed at the works of Mr. J. G. Bodmer of Manchester. It exhibits the same attention to accuracy and completeness which characterizes all the arrangements of that eminent engineer.

The dimensions are given in inches and 32ds of an inch (marked IN.), and in French millimetres (marked M.M.).

Rivets are a species of bolts, usually short, and intended to join two plates together, without nut or screw, being held by forming the end of the pin which passes through the hole into a second head with the hammer. For this purpose they are entered red-hot, and are prevented from receding under the hammer in the process of rivetting by a heavy hammer held against the head. They are exceedingly apt to be injured in the process of rivetting by the action of the hammer, which causes the iron to crystallize often to such an extent that the newly-formed head after a time drops off. To avoid this the rivetting ought to be conducted with as much despatch as possible, and as soon as the head is fairly formed the hammering ought to be discontinued, the injurious effect being produced by the hammer-finishing at a low temperature. The mode of rivetting by pressure, introduced by Mr. Fairbairn of Manchester, is for that reason to be decidedly preferred to that effected with the hammer.

GUIDE PULLEYS.

It frequently happens, when the power is to be conveyed by a belt, that the connection cannot be obtained directly, in consequence of the relative positions of the shafts, which may be placed obliquely to one another; and sometimes other parts of the geering, machinery, cross-beams of the building, and the like, come between the points to be connected, and by their intervention prevent the belt from passing immediately from one point to the other. Under these circumstances, the belt is conducted through a circuitous path to the point required; and not unfrequently it has its plane of action changed in the transit through an angle of even 90°. These conditions are readily fulfilled by directing the belt over a pair of loose pulleys set to the proper angle, in relation to the points of action which the belt is required to connect.

An example of the common guide pulley frame is given in figs. 16 and 17 of Plate II. The axes of the pulleys are adjustable in the frame B to any required angle within a certain range in a common plane, which is parallel with a plane passing through the driving axis and perpendicular to the plane of the driven shaft. When the belt, therefore, passes from the driver over one of the guide pulleys and is returned upon the other, the bight will be thrown into a plane at right angles to the normal plane, and a pulley placed in that plane, its axis coinciding with it, will be driven by the belt without any tendency to change its plane of action.

We have supposed the beams to be parallel and perpendicular to one another; but the arrangement may be adapted to other positions by placing the guide pulleys in different planes; and in order to allow of further accommodation, the frame may be made adjustable at the point of attachment to the beam at D.

TABULAR NOS.	Threads per Inch.	DIAMETER OF SCREW.				DIAMETER OF NUT.		HEIGHT OF NUT.				DIAMETER OF WASHER.				THICKNESS OF WASHER.	
		Top of Thread.		Bottom of Thread.				Low Nut.		High Nut.		Hexagonal Nut.		Square Nut.			
		Inches & 32ds.	M.M.	Inches & 32ds.	M.M.	Inches & 32ds.	M.M.	Inches & 32ds.	M.M.	Inches & 32ds.	M.M.	Inches & 32ds.	M.M.	Inches & 32ds.	M.M.	Inches & 32ds.	M.M.
I.	24	0 6	4.75	0 4	3.17	0 12	9.51	0 9	7.13	0 9	7.13	0 18	14.27	0 20	15.86	0 2	1.58
II.	24	0 7	5.55	0 5	3.96	0 12	9.51	0 9	7.13	0 9	7.13	0 18	14.27	0 20	15.86	0 2	1.58
III.	20	0 8	6.34	0 5	3.96	0 15	11.89	0 9	7.13	0 13	10.30	0 20	15.86	0 24	19.03	0 2	1.58
IV.	20	0 9	7.13	0 6	4.75	0 15	11.89	0 9	7.13	0 13	10.30	0 20	15.86	0 24	19.03	0 2	1.58
V.	18	0 10	7.93	0 7	5.55	0 19	15.16	0 11	8.72	0 16	12.68	0 26	20.61	0 29	22.99	0 2	1.58
VI.	16	0 12	9.51	0 9	7.13	0 19	15.16	0 11	8.72	0 16	12.68	0 26	20.61	0 29	22.99	0 2	1.58
VII.	14	0 14	11.10	0 10	7.93	0 25	19.82	0 15	11.89	0 20	15.86	1 1	26.18	1 6	30.14	0 3	2.37
VIII.	12	0 16	12.68	0 12	9.51	0 25	19.82	0 15	11.89	0 20	15.86	1 1	26.18	1 6	30.14	0 3	2.37
IX.	11	0 18	14.27	0 13	10.30	0 28	22.20	0 19	15.16	0 25	19.82	1 3	27.76	1 10	33.32	0 3	2.37
X.	11	0 20	15.86	0 15	11.89	0 28	22.20	0 19	15.16	0 25	19.82	1 3	27.76	1 10	33.32	0 3	2.37
XI.	10	0 22	17.44	0 16	12.68	1 2	26.97	0 23	18.23	0 30	23.79	1 10	33.32	1 16	38.07	0 3	2.37
XII.	10	0 24	19.03	0 18	14.27	1 2	26.97	0 23	18.23	0 30	23.79	1 10	33.32	1 16	38.07	0 3	2.37
XIII.	9	0 26	20.61	0 19	15.16	1 9	32.52	0 27	21.41	1 4	28.56	1 22	42.83	1 30	49.18	0 4	3.17
XIV.	9	0 28	22.20	0 21	16.65	1 9	32.52	0 27	21.41	1 4	28.56	1 22	42.83	1 30	49.18	0 4	3.17
XV.	8	0 30	23.79	0 23	18.23	1 20	41.25	0 31	24.58	1 10	33.32	2 4	53.96	2 16	63.47	0 5	3.96
XVI.	8	1	25.39	0 25	19.82	1 20	41.25	0 31	24.58	1 10	33.32	2 4	53.96	2 16	63.47	0 5	3.96
XVII.	7	1 4	28.56	0 28	22.20	1 26	46.	1 6	30.14	1 18	39.66	2 10	58.72	2 23	69.02	0 6	4.75
XVIII.	7	1 8	31.73	1 0	25.39	1 26	46.	1 6	30.14	1 18	39.66	2 10	58.72	2 23	69.02	0 6	4.75
XIX.	6	1 12	34.90	1 2	26.97	2 3	53.16	1 14	36.49	1 27	46.80	2 23	69.02	3 4	79.36	0 6	4.75
XX.	6	1 16	38.07	1 6	30.14	2 3	53.16	1 14	36.49	1 27	46.80	2 23	69.02	3 4	79.36	0 6	4.75
XXI.	5	1 20	41.25	1 7	30.94	2 12	60.30	1 22	42.83	2 3	53.16	3 3	78.56	3 19	91.35	0 8	6.34
XXII.	5	1 24	44.42	1 11	34.11	2 12	60.30	1 22	42.83	2 3	53.16	3 3	78.56	3 19	91.35	0 8	6.34
XXIII.	4	1 28	47.59	1 12	34.90	2 24	69.82	1 30	49.18	2 13	61.09	3 16	88.87	4 4	104.76	0 8	6.34
XXIV.	4	2	50.79	1 16	38.07	2 24	69.82	1 30	49.18	2 13	61.09	3 16	88.87	4 4	104.76	0 8	6.34
XXV.	3½	2 8	57.13	1 20	41.25	3 10	84.12	2 12	60.30	2 30	74.58	4 9	108.72	5 0	126.99	0 9	7.13
XXVI.	3⅓	2 16	63.47	1 28	47.59	3 10	84.12	2 12	60.30	2 30	74.58	4 9	108.72	5 0	126.99	0 9	7.13
XXVII.	2⅔	2 24	69.82	2 0	50.79	4	101.5	2 28	72.99	3 18	90.46	5 1	127.78	5 29	149.98	0 10	7.93
XXVIII.	2⅔	3	76.18	2 8	57.13	4	101.5	2 28	72.99	3 18	90.46	5 1	127.78	5 29	149.98	0 10	7.93

ENGINEER AND MACHINIST'S ASSISTANT.

PART THIRD.

ON TOOLS

EMPLOYED IN

ENGINEERING OPERATIONS.

The great and manifest importance of Tools to the mechanic is so self-evident that it is extraordinary the subject has not hitherto received that investigation which it obviously deserves. The vast improvements in modern machinery are mainly attributable to the excellence and accuracy of the tools used in preparing and completing the various parts of which every machine is composed; yet, the only English author who has written specifically on tools is Mr. James Nasmyth of Manchester, to whose excellent and thoroughly practical paper we shall take occasion to refer.

By the expression tools, according to the definition given by Mr. George Rennie,* we understand instruments employed in the manual arts for facilitating mechanical operations by means of percussion, penetration, separation, and abrasion of the substances operated upon, and for all which operations various motions are required to be imparted either to the tool or the work.

For the sake of distinctness it would be desirable, so far as is practicable, to treat the subject under two points of view, 1st. Where motion is given to the tool, as in handicraft work, 2nd. Where motion is given either to the tool or the work, as in self-acting or automatic tools. Now, in the case of the turning-lathe the tool usually remains fixed, whilst the object invariably moves—in that of the planing-machine the tool or cutter may either remain fixed or be made to move, according to the duty required to be performed. In almost all the other machines which come under the denomination *tools*—such, for example, as are intended to perform the various mechanical operations of slotting, key-grooving, punching, drilling, nut-trimming, cutting the teeth of wheels, boring, screw-cutting—the tool receives motion, although in some cases, particularly in the nut-trimming and screw-cutting machines, the tool may be either movable or fixed.

It would afford much matter for curious and instructive inquiry to trace the early history of tools, as there can be little doubt that the use of handicraft tools is coeval with the earliest ages; and assuming the recent researches of modern travellers to be satisfactory proof of the fact that the ancients were acquainted with almost all the tools now in use, we cannot fail to admire the patient perseverance of the workman, whose skill, combined with manual labour, enabled him to produce so many beautiful specimens of his art—a circumstance the more remarkable when we consider the rude and simple implements by the aid of which this extraordinary degree of excellence was attained.

The gradual improvement in tools, which of late years have reached a very high point of perfection, is well illustrated by the Wheel-cutting and Dividing Engines. We therefore propose very briefly to sketch the history of those machines and appliances which come under the general name above prefixed.

While the art of constructing wheel-work was in a less advanced state, the dividing of the circumference of a wheel into the requisite number of parts, and cutting out the tooth spaces by a manual operation, was not only a tedious but also an extremely imperfect way of proceeding. To facilitate such manual operation by a file, the simple platform described by Père Alexandre in his Treatise on Clock-making, was invented; this platform was simply a circular plate of brass, of ten or more inches in diameter, with concentric circles traced thereon corresponding to the numbers of teeth in the wheels and pinions of clock-work. In the centre of this platform was fixed a stud or fast arbor, round which an index, with a straight edge pointing to the centre, turned freely into any given point of a required circle, by means of which the divisions of any given circle were transferred to a wheel placed on the central arbor under the index already described, by a marking point. This mode of dividing a wheel is still practised in some branches of the mechanical arts, and is, doubtless, an easy way of transferring divisions from a larger to a smaller circle for various purposes, where rigid accuracy is not required. But one great difficulty still remained to be surmounted; the spaces necessarily required to be cut by hand with a file. At length a small frame was mounted on the index, which was contrived to direct and confine the file in such a way as to cut the notches in a wheel placed over the index, with less deviation from the truth than could be managed by mere manual dexterity. It is extremely probable that this addition led to the adoption of a circular file or cutter, and of such other appendages as completed the construction of a simple wheel-cutting machine, and it is asserted by M. le Roy† that Dr. Hook was the first person who contrived such an arrangement as could merit the name of a cutting engine. The invention of this eminent mechanician, like many other of his inventions, has proved to be of great and permanent utility in mechanics. It consisted of an entire transmutation of the old stationary platform, with its moveable appendages into a moveable platform inserted into a strong metallic frame with fixed and other suitable appendages. The machine thus converted into a self-acting piece of mechanism was made up of the strong frame, the sliding bars for supporting the platform or plate, with a horizontal screw for adjusting the distance of the circular cutter, the divided plate with a revolving arbor to receive the wheel to be cut, and the *alidad* or index fixed to the great frame in the position of a tangent line to any of the divided circles, and applying its bent and rounded point to any of the pierced marks of division on the circle successively, as the plate revolved, during the operation of cutting the successive teeth of a wheel. This construction of the engine is very nearly identical in principle to that used in the present day, more especially for cutting the teeth of small wheels, as shown in the subjoined cut. Here A shows the arbor on which the wheel to be cut is fixed. B, the cutter. C, a toothed wheel worked by the handle E, and taking into the pinion D, which being on the same axis as the cutter B, imparts to it a velocity proportionate to the number of teeth in the wheel C, and the pinion D. F is a lever handle by means of which the swinging frame may be gradually depressed as the cutter B is brought into operation, or raised when it has performed its work. G, the horizontal screw of adjustment; H, the division plate; and I, the index or pointer.

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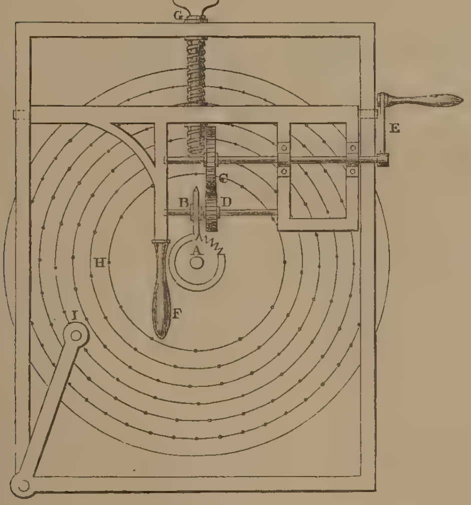
* Buchanan's Practical Essays on Mill Work and other Machinery. Third Edition, with Additions, by George Rennie, Esq., C.E., p. 39.

† Etrennes Chronometriques.

Fig. 1.



The original divisions of the circle, namely, 360, 300, 150, 90, 60, 30, are commonly inserted in the ordinary division-plates, and sometimes even smaller numbers are included. This tends to prevent mistakes in working, but is in other respects superfluous; for, taking every fourth hole in the circle of 360, gives precisely the same result as using the circle of 90; and, similarly, taking each alternate hole in the circle of 300, will be precisely the same in effect as using the circle of 150. It must, we think, be obvious to every one, acquainted with the ordinary process of cutting the teeth of wheels, that engines of the construction just described are very limited in their operations, by reason of their powers extending only to the numbers marked on the divided circles, or the factors of which those numbers are composed, and because the prime numbers are not usually inserted. To remedy this defect, and at the same time render the engine of greater practical utility, appears to have been a favourite study with several ingenious artisans, whose daily avocations admirably qualified them to appreciate the improvements we have already referred to, as well as to devise such additional apparatus as would make the engine more perfect.

It is unnecessary to pursue further the minute details of this subject, but it is to be observed that the only true and accurate method of circular division, namely, by a tangent wheel and endless screw, first contrived and used by Dr. Hooke in 1664, for the purpose of dividing astronomical instruments, has been from time to time advantageously applied to the wheel-cutting engine by several eminent mechanicians both in England and France, among whom we may name Hindley of York, Rehé, mechanist to the Board of Ordnance, and M. Fardoil, whose wheel-cutting engines show an admirable adaptation of the means to the end, and leave little to be desired, although it must be confessed that in some instances the mechanism employed is perhaps too complicated for the plain comprehension of the uneducated workman.*

We now proceed to the Dividing Engine, of which it has been justly observed, that "none has so much contributed to the interest of navigation considered as a science;" indeed the facility, and at the same time the accuracy, with which the measuring portion of any astronomical or mathematical instrument, however portable, can now be divided by our best engines, are truly astonishing; the fine lines of division which in many instances are scarcely visible to the naked eye, are, when magnified by a suitable lens, perceived to be laid down with perfect equality, as to relative distance, so much so, that no one who has not examined the means by which the result is produced, can conceive the possibility that the expedition with which the divisions are made is equal to the accuracy with which they are measured and marked down.

We have already alluded to the method adopted by Dr. Hooke, in 1664, to obtain a perfect division of the circle or a portion thereof. Several trials were made at the Greenwich Observatory, by Flamsteed the Astronomer Royal, but the method was found defective, probably in consequence of imperfect workmanship, and was soon abandoned. In Mr. Smeaton's paper read before the Royal Society of London, November 17th, 1785, on the "Graduation of Astronomical Instruments," he mentions an engine made by Mr. Henry Hindley of York, which indented the edge of any circle in such a way that a screw with fifteen threads acting at once, would, by means of a micrometer, read off any given number of divisions, so as to answer the purpose of subdividing the circle. It does not, however, appear that this engine, though it divided the circles of Hindley's equatorial instruments, was either intended, or indeed so well adapted for graduating circles, as for cutting the teeth of wheels in clockwork. According to the same authority, Mr. Ramsden, in consequence of the reward offered by the Board of Longitude to Mr. Bird, for his method of dividing, in the year 1760, turned his attention towards the contrivance of an engine that would divide nautical instruments with sufficient accuracy, without resorting to the delicate and tedious process of manipulation practised by Mr. Bird. Accordingly, considering the nature and capabilities of the endless screw, and probably contemplating what Dr. Hooke had suggested, and Hindley actually accomplished in this way, though this is merely matter of conjecture, he completed an engine with an indented plate, or wheel of thirty inches diameter, which, though it did not entirely answer his expectations to their full extent, yet was found very useful for dividing theodolites, and such like instruments, with great facility. This was effected before the spring of 1768; and in 1774 a much larger and more efficient engine was produced, with an indented plate of forty-five inches diameter, which divided a sextant for Mr. Bird's examination so accurately, that the Board of Longitude, ever ready to remunerate any successful endeavour to promote the lunar method of determining the longitude at sea, did not hesitate to confer a handsome reward on the inventor, but on the express condition that the said engine should be at the service of the public, and that Mr. Ramsden should publish an explanation of his method of making and using it.*

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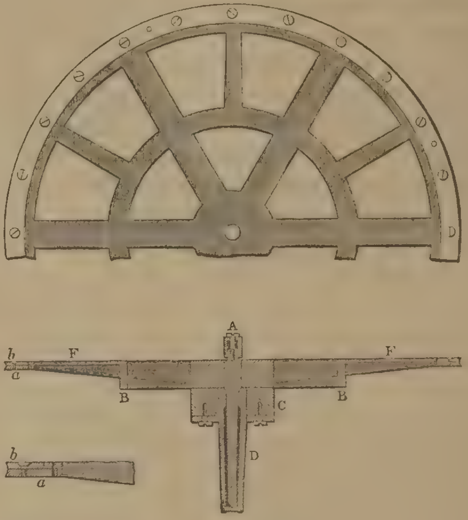
* For more extended information we refer to Berthoud, Essai sur Horlogerie, and his Histoire de la Mesure du Temps. Thiout, Traité d' Horlogerie. Rees's New Cyclopedia, art. Cutting Engine, vol. x. Reid's Treatise on Clock and Watch Making, 2nd edition, p. 283—293, and Emerson's Mechanics, ed. 1825, Note, p. 309.

In 1810 Mr. James Allan was rewarded by the Society of Arts of London, with a Gold Medal; and in 1820, by the Board of Longitude, with one hundred pounds, for his improvement on Ramsden's dividing engine. This improvement consists in the method employed to cut or rack the teeth around the periphery of the great circle, worked by an endless screw, upon which the arc to be divided is placed, so as to ensure perfect equality of size, as regards the teeth, in all parts of the circle. This extremely ingenious, though simple contrivance of Mr. Allan, is described in the transactions of the Society of Arts.† The great circle of bell-metal, a semi-plan of which is shown in fig. 2, is mounted upon an axis A, and its surface made truly plane and perpendicular to the axis; the section shows the figure of the axis, and the central ring B, to give the greatest strength to the circle; C is a section of a portion of the frame of the engine; and D, a socket into which the axis A is fitted. The circumference of the large circle is first turned to such a figure as to receive a ring of brass *a*, which is united firmly to it by a number of pins. Upon this ring a second *b*, is placed, the two making the same thickness as the circle, a sectional view of which is here introduced. The inside of the ring *b*, and the outside of the bell-metal circle, are fitted to each other with the greatest accuracy, and great care is taken to turn the edges truly and concentric with the axis of the circle; the brass rings *a*, and *b*, are held together by twenty-four screws, and a groove corresponding to the curvature of the screw which moves the circle, is then turned in the outside of the two; in this state the racking of the teeth is performed by a screw similar to that afterwards used to turn the circle to its divisions, but notched across the threads so that it cuts like a saw, when pressed against the circle and turned round, and removes the metal from the spaces between the teeth, which are by this means formed around the edge of the circle; when this has been performed all round, two fine lines are drawn across the brass and bell-metal circles, diametrically opposite to each other; the twenty-four brass screws are then withdrawn, and the upper brass ring turned exactly half round, which is determined by the lines before mentioned; and by this means the teeth of the circle are divided into two thicknesses, and being put together again in opposite directions, if any error arose in racking the teeth, it would be shown by the upper and lower halves of the teeth not coinciding when reversed, and by racking them while reversed, the screw would cut away the inequalities, and make all the teeth of the same size and distance from each other; this reversing the teeth is performed several times, till the teeth are brought to a perfect equality in all parts of the circle; four steadying pins are accurately fitted into the two rings to hold them together in any of the positions in which they have been racked together, and it is upon these that dependence is placed for the coincidence of the teeth, the twenty-four screws being merely to hold the rings together, and are fitted rather loosely in their holes that they may not strain the steadying pins.‡

Fig. 2.



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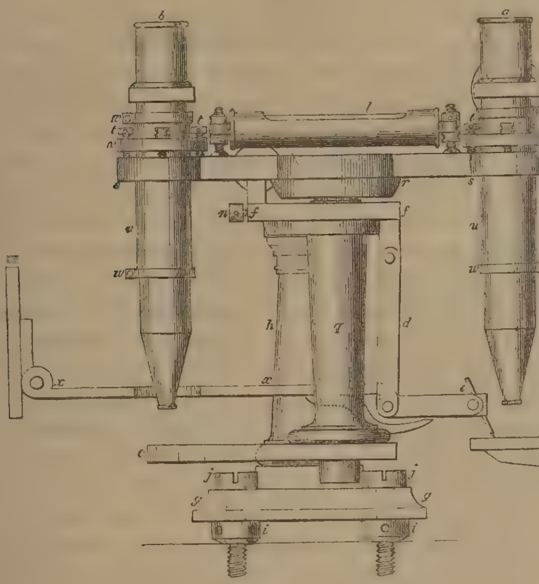
* Published in a 4to pamphlet in the year 1777. The sum of money given to Mr. Ramsden was £615, of which £300 was considered as a reward for his improvement in the art of dividing instruments by means of his engine, and the remaining £315 was paid in consideration of his making over the property of the said engine to the Commissioners of the Board of Longitude for the service of the public.

† Vol. xxviii. p. 179—184.

‡ In 1816 Mr. Allan received the Silver Isis Medal of the Society of

We have purposely omitted any mention of the improved engine by Mr. E. Troughton, in whose hands the art doubtless arrived at a high degree of exactness, because, to adopt the language of a competent judge, there are various difficulties in the application and construction of the apparatus, to avoid which was the avowed object of the engine now in part to be described, by adopting principles perfectly independent of mechanical action, and governed only by vision, assisted by the most powerful optical means. For this really scientific piece of mechanism we are indebted to Mr. Alexander Ross, mathematical instrument maker, to whose ingenuity the Society of Arts of London, in 1831, awarded the Gold Isis Medal and fifty guineas. The annexed cut shows a side view of Mr. Ross's apparatus for cutting original divisions,* and consists of the following parts. A small circle 10 or 12 inches diameter, divided into spaces of 3° 45′ or 96 parts, by the usual dividing engine or by any ordinary means—two micrometer microscopes, represented at *a*, *b*—an arc *c*, of the length of 3° 45′ of the circle to be divided—a cutting frame *d*, *e*, and a frame *f f g g*, to support the apparatus. The frame consists of a bottom and top plate connected by two strong pillars, one of which is represented at *h*, the front one being removed to show the other parts. In the bottom plate are screwed the nuts *i*, *i*, which form adjustable feet for the frame; these nuts are perforated, and the screws *jj* pass through and fasten the whole securely, after being levelled by the nuts *i* and the level *l*. The upper plate is secured to the pillars *h* by two screws and collets, moving on the one as a centre, and adjustable at the other by the pushing screw *n*, for the purpose of setting the cutting point *e*, which is attached to the upper plate *f*, to cut a radiating division on the circle to be divided; an arc and index not capable of being shown in a side view, indicate the inclination given. To the upper plate is likewise attached the hollow centre *q*; in this works a male centre, the flange of which is seen at *r*; this supports the bar *s s*, which carries the microscopes *a*, *b* and the level *l*. The microscopes are secured to this bar by two pulling and two pushing screws *t*, *t*, passing through the flanges *v*, *v*, and acting in and on the bar *s*. The microscopes are secured to the flange by fitting into strong tubes *uu*, and when adjusted to distinct vision, can be fixed in that position by the clamping rings *ww*. The handle *x* for the cutting frame is attached to the perpendicular link *d*, having a double joint at the point where it is fixed, in order to prevent any unequal pressure from producing a lateral motion of the cutting point; the other end is connected to the upright dovetail slide *m*, which forms part of the apparatus for moving the cutting point.†

Fig. 3



Although the object of this essay is not to treat of such minute and delicate instruments as those just described, it should be borne in mind that it is to the refinements introduced into such machines in the first instance that we are, in many cases, indebted for the efficiency and perfection of those more powerful machines which come more immediately under the cognizance of the engineer and millwright.

One of the most valuable aids to the more perfect construction of machinery is due to Mr. Joseph Whitworth of Manchester,‡ who has recently introduced the simple process of *scraping* instead of the dirty and unsatisfactory operation of grinding, as a means of producing plane metallic surfaces. It is generally of importance when such surfaces are placed in contact, that all the bearing points should be in the same plane, that they should be equidistant from each other, and that they should be sufficiently numerous for the particular application intended. Where surfaces remain together in fixed contact, the bearing points may without disadvantage, be fewer in number, and consequently wider apart; but in the case of sliding surfaces the points should be numerous, and in close approximation.

A little consideration will make it evident that these conditions cannot be obtained by the process of grinding. And, first, with regard to general outline, how is the original error to be got rid of? For if it be supposed that one of the surfaces is concave, and the other a true plane, then the tendency of grinding, no doubt, would be to reduce the error of the former, but the opposite error would at the same time be created in the true surface. The only case in which an

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Arts of London for his method of "making screws for mathematical purposes," more particularly the Dividing Engine, and upon which its value mainly depends. (See Trans. Soc. Arts, vol. xxxiv. p. 207—213,) and in 1821, the large Silver Medal of the same Society for his method of "cutting the Divisions on circular Arcs." This contrivance refers, 1st. to the apparatus attached to the cutter, by which the necessity of bounding arcs for regulating the length of the divisional lines, is avoided; and 2nd. to the apparatus for stopping the tangent screw, so as to prevent the possibility of any over-running of the teeth composing the circular rack, and therefore of obviating any inequality from this source in the divisions cut by the engine.

* A full description of Mr. Ross's complete engine is given in Trans. Soc. Arts, vol. xlviii. p. 302—332.

† See further Rees' New Cyclopedia, art. Engine, vol. xiii.; and art. Graduation, vol. xvi.

‡ On Plane Metallic Surfaces and the Proper Mode of Preparing them, read at the Meeting of the British Association at Glasgow, A. D. 1840. By Joseph Whitworth. Privately printed.

original error could be extirpated, would be when it was met by a corresponding and contrary error of exactly the same amount in the opposed surface; but it is evident that where only two surfaces are concerned, the variety of error in the general outline is not sufficient to afford any probability of mutual compensation. It will farther appear, that if the original error be inconsiderable, the surfaces must lose instead of gaining truth. It results from the nature of the process that certain parts are acted upon for a longer time than others; they are consequently more worn, and the surfaces are made hollow; nor is there any probability of obviating this source of error except by sliding the one surface entirely out of the other at each move, a method which is clearly impracticable.

It may be mentioned as an additional cause of error, that the grinding powder collects in greater quantity about the edges of the metal than upon the interior parts, producing the effect of excessive abrasion of the former. This is particularly objectionable in the case of slides, from the access afforded to particles of dust, and the immediate injury necessarily occasioned thereby. Another circumstance materially affecting the durability of ground slides, is that a portion of the emery becomes fixed in the pores of the metal, and can never be entirely eradicated therefrom, causing a rapid and irregular wear of the surface.

If then grinding be not adapted to form a true general outline, neither is it to produce accuracy in the minuter detail. There can be little chance of a multitude of points being brought to bear, and distributed equally under a process from which all particular management is obviously excluded. To obtain any such result, it is necessary to possess the means of operating independently on each point as occasion may require; whereas grinding affects all simultaneously. It is subject neither to observation nor control, there is no opportunity of regulating the distribution of the powder, or of modifying its application, with reference to the particular condition of the different parts of the surface. The variation in the quantity of the powder and the quality of the metal, will of necessity produce inequalities, even supposing they did not previously exist. Hence, if a ground surface be carefully examined, the bearing points will be found lying together in irregular masses, with extensive cavities intervening. An appearance indeed of beautiful regularity is produced, to which, no doubt, we may trace the universal prejudice so long established in favour of the process; but this appearance, so far from being any evidence of truth, serves only to conceal error, and under this specious disguise surfaces pass without examination, which if unground would be at once rejected.

In addition to what has been stated, it must be remembered another great evil of grinding is that it takes from the mechanic all sense of responsibility and all spirit of emulation, while it deludes him with the idea that the surface will be ultimately ground true; hence he slurs his work over in a slovenly manner, trusting to the effect of grinding, being conscious that it will efface all evidence either of care or neglect on his part.

Thus it appears that the practice of grinding has altogether impeded the progress of improvement. A true surface instead of being a thing of frequent occurrence, was until lately almost unknown; few mechanics have any distinct knowledge of the method to be pursued for obtaining it, nor do practical men sufficiently appreciate either the immense importance, or the comparative facility of the acquisition. The expression "true surface" used in reference to metallic bodies may require qualification. Absolute truth is confessedly unattainable; moreover it would be possible to aim at a degree of perfection far beyond the necessity of the particular case, the difficulty of which would more than counterbalance the advantage; nevertheless it is certain that the progress hitherto made falls far short of this practical limit, and that considerations of economy alone would carry improvement many degrees higher. The extensive class of machinery denominated tools, affords an important application of the subject; here every consideration combines to enforce accuracy. It is implied in the very name of the planing engine, the express purpose of which is to produce true surfaces, and it is itself constructed of slides, according to the truth of which will be that of the work performed; and when it is considered that the lathe and the planing engine are employed in the making of all other machines, and are continually re-producing surfaces similar to their own, it will manifestly appear of paramount importance that they should themselves be perfect models. Indeed it would be difficult to mention any description of machinery which would not serve as an illustration of the importance belonging to truth of surface, and at the same time offer abundant evidence of the present necessity for material improvement; nor is there any subject connected with mechanics, the bearings of which, whether regarded in a manufacturing or scientific point of view, are more varied or more extensive.

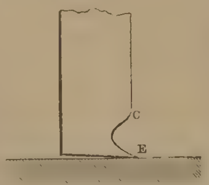
The tool employed for scraping is not only simple but easily made; it should be of the best cast-steel, and carefully sharpened to a fine edge on a Turkey-stone, the use of which must be frequently repeated; but worn out files may be converted into convenient scraping-tools. A flat file with the broad end bent and sharpened will be most suitable in the first instance, and afterwards a three-angled file sharpened on all the edges. The process of scraping is equally simple, requiring rather care than skill on the part of the workman, whilst it affords a certain and speedy means of attaining any degree of truth that may be deemed necessary, thus tending to the gradual establishment of a higher standard of excellence, the influence of which cannot fail to affect beneficially all mechanical operations, opening at the same time to the mechanic himself a new field in which he will find ample scope for the exercise of skill both manual and mental.

We are now in a condition to proceed with the matter more immediately under consideration. The value of every cutting instrument depends upon the excellence of the steel of which it is made, the care bestowed during the several processes of forging, hardening, and tempering, and the just adaptation of the angle or bevil which forms its edge to the work it is intended to perform. Generally speaking, this angle is determined by the hardness of the substance to be operated upon. Thus we see chisels for cutting soft woods are thinner than those used for the harder species, and these again are more acute than chisels employed for cutting metals; or in other words, the greater the resistance offered by the material to be cut, the more obtuse must be the angle of the tool. This definition is not propounded as rigidly correct in all cases, although it is susceptible of abundant practical illustration; for example, in hand-turning the workman is enabled by raising or lowering the T of the rest, to vary the direction and limit the cut of the tool employed, according to circumstances. This one fact, amongst a multitude of others equally palpable that could be adduced, might have been expected to induce inquiry and investigation. On the contrary, we have the authority of Mr. Nasmyth* for stating, that the form of tools, more especially those used in turning and planing iron, brass, &c., has not hitherto received either that attention which the importance of the subject calls for, nor has any attempt been made to reduce it to plain and general principles, of which it is highly susceptible, and if so treated would be of much service to those in whose hands the management of such tools is for the most part entrusted. So many considerations of a practical nature are inseparable from this subject, that the quality as well as the quantity of work produceable from turning lathes and planing machines depends entirely upon the skill of the workman in giving to his tools the proper form.

The general principle propounded by Mr. Nasmyth, which is equally applicable whether the motion be horizontal, circular, or vertical, is deduced from a consideration of the direction in which the metal is to be cut or penetrated. With regard to the first case, as in the planing machine,† it is manifest that the face of the tool is at right angles to the plane of the material to be cut, and consequently, if its point or cutting edge be made in the form of a very obtuse angle it will possess little or no penetrating quality—such a tool would not cut, but rather abrade or probably crush off the particles of metal. Again, if we resort to the other extreme, and give to the cutting edge the shape of an extremely acute angle, we shall find, however *sharp* it may appear, a total absence of penetrating quality, or at all events in the required direction, and what is equally objectionable, the point being weak would snap off, incapable of resisting the least applied force.

From an investigation of these and other obvious facts, Mr. Nasmyth concludes that a tool of the form shown in the annexed cut fulfils the requisite conditions, as it combines a high degree of acuteness with sufficient strength—the former in the direction of the cut, and the latter behind the point or cutting edge, where it is most needed. Hence the following principle may be established—namely, that in forming and setting a tool to cut any surface, it is essentially necessary so to place it that *the end shall form the least possible angle with the surface to be cut*, or in other words, as nearly parallel as possible, and whatever degree of acuteness may be deemed necessary must be obtained by hollowing out the face E C, on which the shavings slide.‡

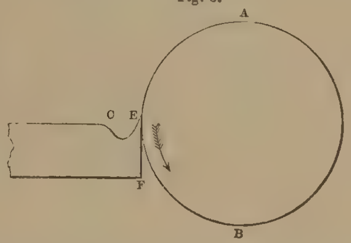
Fig. 4.



The same principle is similarly true in the case of turning-tools, and indeed, in every tool, from the smallest and most delicate of the clock and watch maker, up to the largest and most powerful tool in an engineer's lathe or planing machine.

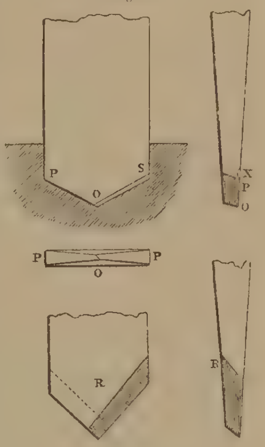
As regards circular motion we have a clear exemplification of the principle by merely considering the tool already described as a turning-tool. Here A B shows a section of a cylindrical bar in the lathe, and E F so placed as to be as nearly as circumstances will permit a tangent, that is, at right angles to the radius of the curve—the requisite acuteness being obtained, as before, by hollowing out the face E C.

Fig. 5.



The same principle applies to drills. Thus fig. 6, denoting the end of a drill, the edge O P should be in the least degree prominent or out of the plane of the surface, of which the bounding lines are the edges; O S being slightly less prominent than O P, so that the penetrating quality at the edge O P may be limited as much as possible. An adherence to these rules will produce a drill that shall cut a smooth and equal hole, without chattering, as is commonly the case when the edges are bevilled very much back, as shown at R. The necessary acuteness to the cutting edge of the drill is easily obtained by merely observing the principle laid down in respect to turning tools, that is, by hollowing out a groove at X, on each cutting face.

Fig. 6.



Every mechanic is sensible of the value of good tools as necessary appliances to the performance of his work more quickly, with less exertion, and more accurately than can be done with inferior ones; yet how few are in a position to answer this apparently simple question, what constitutes this quality denominated *goodness?* The excellence of cutting tools is generally decided by their relative degrees of endurance, but how many incidental circumstances may, and frequently do, interfere to vitiate any accurate comparison! As regards hardness, nearly the only test is the resistance the objects offer to the file, a mode extremely fallacious, because files differ among themselves in hardness, and at best only serve to indicate in a very imperfect manner, to the touch of the individual, a vague notion without any distinct measure. Take for example two chisels for turning iron, both of cast steel, and from the hands of the same maker, and although precisely alike in outward appearance, the one may be absolutely worthless and the other of high quality. Nay, one portion of the same chisel may be good and the other bad. If during the process of fabrication of two cutting tools the same treatment be practised with similar care, we should naturally expect, all things being coincident, that the one would correspond with the other. Experience shows the fallacy of this mode of reasoning. Nearly every metal turner makes his own tools, and for this plain reason, that he cannot place any dependance on those he purchases. The smith who forges, hardens and tempers these tools, rarely uses them; he labours to produce a certain number in a given time, and if they satisfy the eye his object is attained. The good or bad quality of a tool depends more on the care and attention bestowed during the process of forging than is commonly imagined, and the defects, whether of texture or edge, which so often present themselves in articles manufactured of steel, are to be traced not so much to any natural imperfection or partial conversion of the metal as to a slovenly and hasty mode of forging.

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* Buchanan on Mill-work and other Machinery, by Rennie. Appendix B.

† It is immaterial, so far as the principle here contended for is concerned, whether the tool be stationary and the table of the machine moveable or the reverse—since the action is precisely the same in either case.

‡ An apt and very familiar illustration of the principle may be drawn from the common plane of the joiner. An artificial end being given to the plane-iron, which is here the cutting tool, by means of the sole of the plane, this necessarily limits the penetrating quality in all directions except that in which it is required to remove the material. Further, it can scarcely have escaped observation that the bevilled surface of a chisel is invariably placed outwards, and the flat surface next to the wood, so that the face of the chisel next the wood and the surface of the wood itself shall form the least possible angle.

In order to elucidate a subject confessedly of the highest possible importance, but unfortunately obscured beyond any other connected with mechanical art by conflicting opinions, we now propose to investigate the nature and properties of steel, and to introduce a variety of practical examples collected from authentic sources, in preference to any speculative theories. These examples will be found in most cases to confirm, but in some to confute each other, thus leaving to each individual the means of testing by his own experience the value of the information offered.

Steel, as is well known, is made by combining carbon with iron, the atmosphere being excluded and a white heat kept up until the iron has imbibed from the carbonaceous matter with which it is surrounded a sufficient quantity, which may be more or less according to the use for which the steel is intended. Iron is very slightly, and if pure not at all, altered or increased in hardness by sudden cooling from a red heat, but the small amount of carbon which it receives during the process of cementation greatly increases both its strength and toughness, leaving it alike malleable and ductile, and imparts to it that peculiarly valuable property of becoming extremely hard if suddenly cooled from a red heat. With this first dose of carbon it is denominated *mild steel*, possessing all the distinctive properties of iron with increased strength. A larger dose of carbon renders the metal susceptible of greater hardness, and proportionably more brittle. It is also fusible, and therefore called *cast steel*, but being less malleable is more difficult to work.

Steel made by cementation is designated *blistered steel*, because it is supposed, while the carbon is entering it meets with oxygen, hydrogen, or some foreign matter which it causes to become gaseous, and thus blisters the surface of the steel. Dr. Thomson* attributes these blisters to a gas evolved in the interior of the bar, which pushes up by its elasticity a film of the metal, and Mr. Gill † considers them as indications of the quality of the steel, as "the hardest will be found to be blistered all over its surface, while the milder will be smoother." Cast steel being made by fusion admits of an equal distribution of the carbon, to the expulsion of every other substance, which cannot endure the intense heat: the soundness of this description of steel is obviously a great recommendation, but the excess of carbon renders it harsh and consequently intractable; under the hammer, however, by careful treatment during the operation of forging, the excess of carbon may be dissipated and the quality of the steel ameliorated and greatly improved for general purposes.

The question whether steel contains any thing besides iron and carbon is purely chemical, the consideration of which would form, did space allow, an interesting theoretical illustration of the practical details of the present inquiry.‡

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* Chemistry of Inorganic Bodies, vol. i. p. 496.

† Technical Repository, vol. i. p. 136.

‡ Steel appears to occupy an intermediate place between cast and malleable iron. The researches of the French Academicians, Monge, Berthollet and Vandermonde, show the distinction between cast iron and steel to be that the former is charged with a superabundant, the latter with a minute yet sufficient dose of carbon; hammered iron, on the contrary, if pure, consists of iron free from all heterogeneous matter. It is to be regretted that the constituent proportions of steel have not been accurately determined. Vauquelin assumes the average amount of carbon to be 1-150th, and Clouet places it as high as 1-32nd. Mr. Parkinson considers the quantity of carbon necessary for making of steel to be very small, indeed the actual amount seldom exceeding 1-200th, or 1-300th, and perhaps never more than 1-100th, the remaining portion of charcoal flying off at the time of cementation in the form of gaseous oxide of carbon. Dr. Thomson analysed some specimens of cast steel, from the manufactory of Mr. Buttery near Glasgow, and the general results of his trials gave the constituents as follows ;—

Iron, - - - - - - - -	99
Carbon, with some Silicon, - -	1
	100

Now this approaches—

Iron, 20 atoms, - - - - -	70
Carbon, 1 atom, - - - - -	0.75
	70.75

And this Dr. Thomson considers as likely to be the constitution of cast steel. He did not in like manner attempt the analysis of blistered steel, but concludes the proportion of carbon in it to be rather less. It is well ascertained that iron and carbon are capable of combining together in a variety of different proportions: when the carbon exceeds, the compound is carburet of iron; when the iron exceeds, the compound is steel or cast iron in various states according to the proportion: all these compounds may be considered as subcarburets of iron. The most complete detail of experiments on these compounds which has yet appeared in this country is by Mr. Mushet. This ingenious metallurgical chemist has observed that the hardness of iron increases with the proportion of charcoal with which it combines, till the carbon amounts to about

A good workman merely requires steel free from flaws, completely homogeneous, and such as will harden at the lowest heat, for this test supersedes all others in proving its superior quality.

Perfectly pure iron cemented in equally pure carbon would doubtless produce steel free from blisters, but as in practice these blisters are unavoidably evolved, it is needless to inquire into their origin more minutely than we have already done, especially as it seems to be admitted that blistered steel is unequally carbonized, the outside retaining the larger portion. It is therefore rendered fit for the market by doubling and welding several times, by which means the parts are more intimately blended together, and the carbon more equally distributed; in this state it is called *sheer steel.* These repeated weldings, although they tend to condense the metal, are apt to produce flaws, 1st. by imperfect union, 2d. by the carbon burning out of the commingling surfaces, thereby interposing a stratum of iron or imperfectly converted steel, and this being softer than the surrounding particles would give way during the extension of the steel. To whatever cause such defects are to be attributed must necessarily remain a matter of conjecture, but that they do very largely accompany this description of steel is certain, and it is a question whether any process short of actual fusion can totally remove them; nevertheless, it is ascertained that long continued forging essentially conduces to the soundness or homogeneity.

Besides these flaws there is another obstacle frequently met with in steel: it is said to have *pins*, when in the operation of turning or filing, knots are developed harder than the other portions of the metal; these knots or pins present themselves of almost every degree of hardness, commencing with mere harshness, and advancing to absolute intractability, so that whilst turning in the lathe the pins would remain projecting out and grind or break the edge of the tool rather than submit to be cut away, and it is by no means unusual to find their hardness nearly approach that of a file applied to remove them. Various causes have been assigned for these knots; Mr. Varley thinks they are portions of metal over-steeled, that is, so completely charged with carbon as to be incapable of being annealed by any known process of slow cooling. Mr. Clement states that he broke the steel across these pins, having filed away the back to render it weak enough to part at the right place, when he found a cut or division, on which account he attributes the flaw and its extreme hardness to an oxide of iron, which prevented the union of the parts. It would be a curious and by no means an unprofitable investigation to analyse the condition of the deepest blisters, in order to determine whether they are alloyed or oxidised, or in any way differing in their state of carbonization from the more solid parts. It seems clear that if these pins are induced by the presence of oxygen, then the adjoining metal would be iron, for there would be a gradation from the oxide through iron to the steel, and consequently the circumference of such a spot would be softest.

An excess of carbon renders steel harder and more brittle, therefore an inequality is liable to occur. This may be illustrated by the known fact, that portions of an iron casting intended to be soft are frequently hardened by contact with the moist sand, of which the mould is formed, and those parts nearest the outside break with a fracture more glassy than even hard steel. Now good steel hardened by sudden immersion in cold water, when at a red heat, will invariably return to a soft state by slow cooling from such heat, and more equally so if the external atmosphere be carefully excluded; but this hard cast iron on the contrary does not; it requires to be exposed for many hours to an intense heat, and must not be smothered by fuel to prevent the escape of the superabundant carbon with which it is charged. The air too should be allowed free access as a means of disengaging some portions of the carbon, while the remainder has a tendency to equalise itself; then, if slowly cooled, the mass will be found to be sufficiently annealed.

The knots or pins in steel are rarely removed by slow cooling alone; there is, however, an opinion prevalent among workmen that *pinny* steel may be rendered uniform in its substance if it be first hardened and then annealed. To burn out these pins would manifestly spoil the steel, because it has no carbon to spare but in the pins (supposing this to be the cause of their hardness), and the process of annealing in air-tight vessels is not found to produce equality sufficient for any good purpose. Even cast steel, which is undeniably purer and more homogeneous than any other description, is liable to long streaks or veins harder than the other portions of the bar. All these show the necessity of greater and more minute attention to the treatment of steel than the subject appears to have received, and for this end two modes of hammering are indispensable. To illustrate our position, let us take any article forged in the usual way, not out of blistered or sheer steel, both of which may be presumed to contain carbon unequally distributed, but of cast steel, which having been fused and passed through the rolls, or under a ponderous tilt-hammer, is characterized as refined and considered uniform. Yet notwithstanding every precaution, we

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1-80th of the whole mass. The hardness is then a maximum, the metal acquires the colour of silver, loses its granulated appearance and assumes a crystallized form. If more carbon be added to the compound the hardness diminishes in proportion to the quantity, as appears from the following tabular arrangement extracted from Mr. Mushet's papers on Iron and Steel.

Iron semi-steelified,	contains	1-150th	of carbon.
Soft steel, capable of welding,	...	1-120th	...
Cast steel for common purposes,	...	1-100th	...
Cast steel requiring more hardness,	...	1-90th	...
Steel capable of standing a few blows, but quite unfit for drawing,	...	1-50th	...
First approach to a steely granulated fracture,	...	1-30th 1-40th	...
White cast iron,	...	1-25th	...
Mottled cast iron,	...	1-20th	...
Carbonated cast iron,	...	1-15th	...
Super-carbonated crude iron,	...	1-12th	...

Dr. Schafthacutl has lately propounded a novel and startling theory, viz. that steel is entirely a mechanical production of the forge-hammer, which tears the molecules of certain species of white cast iron out of their original positions, into which the forces of attraction, in respect to the centres as well as to the position of the molecules, had arranged those molecules by the slow action of heat, and that steel, as it comes out of the converting-furnace or the crucible, is nothing more or less than white cast iron, of which Indian steel, called *wootz,* is the fairest specimen.

still see the labour and skill of the machinist defeated by those veins, fissures, and pins, which denote either metal of inferior quality, or that the original texture of the steel has been deteriorated by the ignorance and carelessness of the smith. The first supposition does not apply to cast steel, in which the dose of carbon is diffused equally throughout the mass, or so nearly so as to render the difference inappreciable. We are therefore compelled to admit the second position, and this, unless we have entirely mistaken the bearings of the case, will enable us to account for past failures and guard against future disappointment.

The twofold process of hammering already alluded to is intended to correct the greater part, if not the whole, of the defects pointed out. It is necessary in the first place to hammer the steel at a forging heat, so as to knead the parts together and keep them moving among themselves. This should be continued till the different constituents are not only intimately blended, but as it were dissolved in each other, so as to insure perfect uniformity, for the carbon being thus spread the metal will be rendered as sound as can be expected of cemented steel; and it is clear that if by mechanical agency all foreign matter be expelled and the carbon alone remain, there is nothing to prevent a perfect union of the parts while under the hammer. Good steel consists of that proportion of carbon and iron, which combined form the strongest and toughest compound; each purer portion, therefore, when brought into contact by the hammer remains in that state and resists its percussive force the more from the greater cohesion of the particles. Hence the redundant or deficient portions suffer most till they become equalized, and the impurities are either beaten out or formed into a homogeneal compound with the entire mass.

Although by this means sound steel may be obtained, it is far from being in a perfect state; it is still very unequal in density, and in a state of distraction; some portions are close and dense, and others are fissured. A second hammering at a particular heat is therefore necessary, and under circumstances required by the shape of the steel—such as recesses in the anvil or blocks laid thereon, technically termed *swadges* and *moulds*. For this purpose the metal is first brought as near as eligible to the required dimensions, and is then to be hammered in order to close and condense the particles equally and throughout, yet leaving every part in a state of rest and ease—a condition very essential for good springs, and indeed, every article formed of steel, that has to vibrate or act by tension.

This second hammering is also intended to prepare the steel for receiving the utmost hardness of which it is susceptible—a quality which entitles it to be considered the *master metal*—the one by which we give shape and form to all others. Now steel at a red heat, when suddenly plunged in cold water, becomes both brittle and hard, but even in this state its toughness greatly exceeds that of any other brittle substance. This characteristic hardness cannot be given in part, but always in full and to its highest limit. So true is this, that in a piece of steel, a portion of which is hard and a portion soft, no gradation of hardness can be detected, the parts adjacent to the hard portion being quite soft, or, as some think, softer than if slowly cooled. This singular fact has been thus accounted for:—Suppose a rod of well-hammered steel to be heated at one end for hardening, there will be a gradation of temperature from the coldest to the hottest extremity, and the annealing or reduction of that hardness which it has received will be in proportion to the heat, consequently the rod will be softer and softer towards the end where the heat is applied. On plunging the bar into cold water, that portion which has become sufficiently hot to harden, is rendered quite hard, but that part immediately adjacent to it will be found to be most annealed, and will endure more twisting and bending than any other. Although this hardness may be imparted in its full extent, it may nevertheless be lowered in any assignable degreee—that is, a portion of its brittleness may be removed by the application of moderate heat, a greater portion by more heat, and so on, as the purposes may require. This is called *tempering*. If hard steel be brought to a red heat and then suffered to cool slowly, it will become as soft as if never hardened: this is called *softening*, and is distinguished from *annealing*, which is a similar process of slow cooling, but applied to steel, iron, or brass, merely to remove all mechanical condensation, whether by hammering or otherwise; for if metal has been altered in shape by the hammer or any other process, as much as it will bear without breaking, then by annealing it will be softened and may again be altered in form as often as requisite. Now as different degrees of heat remove different degrees of condensation received from the hammer, and a white heat removes all, it is of great importance to harden steel from the lowest possible degree of heat, in order to retain as much condensation as practicable; and it is a fortunate coincidence that the greater the condensation the lower is the heat from which steel will harden, and the stronger and tougher it will be. But should this condensed metal be once over-heated, it will then no longer harden from that lower degree, but only from a heat nearly approaching that to which it was originally raised. In this case, the condensation, with all its attendant advantages, can only be restored by re-hammering.

The lowest heat at which steel will generally harden, is a dull or cherry-red, just visible in day light; therefore, to be safe, the same test, that is, a dull red heat, just perceptible in the dark, is chosen for the process of hammering; it offers, too, the advantage of coating the article with carbonaceous matter, thereby securing, instead of losing by the action of the fire, a due supply of carbon, which is of particular consequence. Different modes of performing this part of the process may be adopted. It is desirable that the forge should not be under the influence of a strong light; the anvil should be placed as near as circumstances will permit to the flat bed of the forge, and the fire smothered with small fuel, just kept alive by the bellows, so as never to allow the gas to burst into flame. The pieces of steel under treatment are now placed in the smouldering and partially kindled fuel enveloped in smoke, whence they imbibe a portion of carbon, which the hammering heat is insufficient to expel; they are then brought in succession from the fire to the anvil, and back again to the fire when too cool; the hammer is moved quickly, and every part of the steel subjected to its blow. The position of the article is then slightly changed, and the operation continued and repeated as often as needful, till it has been well hammered in every direction. Experience will soon teach the workman the art of discerning, both by the sound of the blow and the feel consequent to the decreased effect produced by the hammer, when the proposed end is attained and the steel sufficiently condensed. Although at this comparatively low heat the metal spreads much less than it does under the influence of a higher temperature, it must be remembered that unless it be confined in proper recesses or moulds, the expansion produced by the inner acting on the outer portion will be so considerable as to render the exterior band worthless.

From what has been stated, it seems clear that there are two principal causes for the failure of steel, particularly if used quite hard. The first is an unequal combination of carbon with the iron, and this is more or less the case with all steel till properly hammered; the toughness of some parts, the brittleness of others, and the different states of tension hence arising, cause the metal to be liable to crack, if not in hardening yet subsequently, when under the pressure of a severe strain, or the accumulating force of a descending blow. The second cause is bad or imperfect hammering, for it is undeniable, that however much hammered steel may be, it is often left by the workman in a violently conflicting condition, some parts girded and compressed, whilst others are as much strained almost to breaking.

Steel well and duly hammered requires the least tempering to produce the necessary degree of toughness, but when hardened at too great a heat it loses all the previous condensation from the hammer, and with it so much strength that the tenacity disappears, and the same hardness with less strength shows itself in brittleness. Steel, therefore, thus incautiously treated, requires more letting down by tempering to obtain a practical degree of toughness. Such tools are far too weak and soft for turning steel or iron, and do not stand long for any purpose. It seems feasible to imagine that there is a given degree of cold to which steel must be brought in a given time to cause hardness; also a given degree below which it will not harden, and that any further increase of heat only serves to weaken the metal, as likewise any further increase of cold only tends to harden a greater mass; or a deeper coat of a large mass by cooling it in the required time. Steel much overcharged with carbon is too harsh or brittle to receive all the improvement that hammering would otherwise impart; but by selecting the most malleable steel of sound quality, and hammering it at a heat so low as may allow it to retain a coat of carbonaceous matter, it will imbibe the carbon so slowly and in such minute quantities during each hammering, as to enable the workman to enforce the fullest charge compatible with sound hammering. To this limit carbon must improve the strength, but beyond it brittleness comes on, and it refuses to receive any condensation from the hammer, by which alone toughness is given. It is obviously useless to endeavour to obtain hardness if unaccompanied by toughness, as we should be compelled to resort to extreme letting down in tempering to avoid breaking; yet many suppose the hardness to be usefully increased by coming up to this brittleness; therefore, taps, dies, and some turning-tools, in order to combine extreme hardness with toughness, are allowed to imbibe a little more carbon on their surface by placing them in red-hot carbonaceous matter for hardening them, while the tenacity of the interior metal remains unaltered. For this purpose animal charcoal is generally used—burnt leather answers the object, and appears to impart the carbon much quicker than wood charcoal.*

In conducting the actual process of forging it is important to remember, that steel being composed of iron and carbon, the latter substance becomes *volatile* at a high temperature. When too strongly heated, that is to say, when the carbon is volatilized, steel loses its characteristic properties, and assimilates itself to iron; consequently, instruments made of it cannot be expected to possess either keenness or durability of edge. In working cast steel more especially, it is impossible to use too much precaution, since, although some workmen are of opinion that overheated steel may be restored, and even carry prejudice so far as to prefer this recovered steel for cutting-tools, it is a fact that there is no *certain remedy* for the evil.† It cannot, therefore, be too strongly urged, that steel should be worked with the *fewest number* of heats, and at the *lowest temperature* possible to forge it, notwithstanding the additional labour and time expended. In a forge with a naked fire it is a matter of difficulty to regulate the heat, especially with thin or small articles. With these a *muffle* becomes a most useful auxiliary towards attaining and continuing the proper temperature By the muffle the steel is prevented from immediate contact with the coals, as it is in the common forge fire. A piece of gun-barrel, one end being closed, makes an excellent substitute for the muffle, and has the advantage of being less liable to crack or open than either a cast iron or earthen muffle. Pit coal is the fuel most commonly used by smiths, but it is well known that the sulphurous gas disengaged during the combustion of bitumenous coal is extremely prejudicial to steel. For this and other reasons cinders and coke are to be preferred, and the latter is almost invariably selected by the Sheffield workmen, both for forging and hardening.

In hardening and tempering steel three considerations present themselves to our notice:

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* This is clearly a modification of case-hardening. Steel may alike be softened or entirely decarbonized. According to Dr. Ure the common way to soften steel is to put it into an iron case, surrounded with a paste made of lime, ox gall, and a little nitre and water, then to expose the case to a slow fire, which is gradually increased to a considerable heat and afterwards allowed to go out, when the steel is found to be quite soft.—*Dictionary of Arts and Manufactures.* Mr. Jacob Perkins first decarbonizes the metal, and then reimparts the proper dose of carbon by a process similar to case-hardening. A very full and interesting account of the practical details will be found in *Trans. Soc. of Arts,* vol. xxxviii.

† Cast steel injured by over-heating may be partially recovered by four or five re-heatings and quenchings in water, each carried to an extent progressively a little less than the first excess, and lastly, the steel must have a good hammering at the ordinary red heat. It may, however, be doubted whether any degree of hammering—in this process evidently the principal requisite—will restore the loss of tenacity consequent upon the over-heating, or even the too frequent heating of steel. See Holtzapffel's Mechanical Manipulation, vol. i. p. 230.

I. *The means of communicating the necessary degree of heat.*

II. *The means of rapidly abstracting the heat.*

III. *The means of applying a partial or reduced heat for tempering or letting down.*

Small tools, such as watchmakers' drills, are heated in the blue part of the flame of a candle, and are occasionally supported upon charcoal; drills of somewhat larger size are heated with the blow-pipe flame directed very obliquely and a little below the point; but for articles that are either too large to be heated by the blow-pipe or too small to be conveniently exposed to the naked fire, and indeed, in all cases where it is desired to harden tools of any considerable length, it is prudent to adopt some protective contrivance, so that the requisite heat shall be equally and uniformly diffused. Thus, an iron tube or sheet-iron box offers a safe and easy way of operating.

Various methods have been suggested in order to obtain a perfectly uniform temperature in hardening long and thin articles, such as gravers for artists, and other delicate instruments. Melted metal, the fusing point of which is just below the right temperature, has been recommended, and if the melted metal be in sufficient quantity and the heat accurately measured by a pyrometer, the result must prove satisfactory. We are indebted to Mr. Nicholson for a simple yet effectual plan by which the difficulty experienced in communicating to some articles a uniform degree of heat in an ordinary fire is completely obviated. A bath of melted lead, that is, fine lead, containing little or no admixture of tin, is heated to moderate redness and well stirred; into this the steel is plunged for a few seconds, until, when brought near to the surface, that part does not appear less luminous than the rest; the steel is then quickly stirred in the bath, suddenly drawn out, and plunged into a large body of water. In this manner a plate of steel may be hardened so as to be quite brittle, and yet continue so sound as to ring like a bell, "an effect," observes Mr. Nicholson, "which I could never procure in any other way."* Some workmen substitute a pan of charcoal dust heated to redness for the same purpose.

A large proportion of tools in general use are heated in the common forge fire, which should consist of cinders, coke, or charcoal, rather than fresh coal. Recourse is also had to hollow fires, but in all cases the blast from the bellows should be sparingly applied, and principally limited to getting up the fire before introducing the work, which must be allowed sufficient time to get hot, or as it is technically expressed, to "*soak.*"

But whatever method be adopted for heating the work, it is very important that a uniform temperature be communicated to all the parts requiring to be hardened, and this can only be insured by cautiously moving the work to and fro, so as to subject all parts equally to the action of the fire. The safest rule is to harden as little as possible, above the state intended to be produced by tempering, and it is obviously better to err on the side of deficiency than of excess of heat; the definite point is rather critical, and necessarily varies with different qualities of steel; therefore, until the character of the steel be well ascertained, (and this knowledge must result from actual trial†), it is prudent to exercise due

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* Martin's Circle of the Mechanical Arts, 4to, p. 261, 262.

† This point involves two considerations:—1st, The quality of the steel; 2nd, The suitable heat.

As it is a rare occurrence to find any two bars even of the best cast steel of similar quality, it is therefore desirable to adopt some method of proof, which may enable the workman to assort the bars according to the indications they respectively exhibit during the process, as it may be presumed that every part of a bar of cast steel will probably turn out of the same quality as the portion subjected to proof, however dissimilar any number of bars themselves may be relatively considered.

To ascertain the quality of cast steel, the bar must be carefully heated at one end, and drawn down to dimensions suitable for bending, that is, about two inches long and half an inch wide; this must then be heated to the proper degree for hardening, and quenched, leaving the thick portion of the bar still hot, then blazed off to spring temper, and again quenched. The proof is thus conducted: About half an inch of the hardened and tempered portion of the bar is screwed firmly, and in a horizontal position, between the chaps of a strong vice; the operator then, using the bar as a lever, walks round the vice, and gradually bends the steel; if it snap suddenly on being but little strained, it may be pronounced *hard;* if it endure bending a *quarter of a circle,* and then breaks quietly, it is of *mild* or middling quality; but if it bear to be carried round a full *semicircle,* and then separates like lead, it is a proof of *softness,* and well adapted for springs.

With regard to the heat, it is quite impossible to define the exact limit. Cast steel should never be heated above *cherry-red,* just visible in daylight, but other varieties are generally so impure and unequally carbonized, that the degree of heat which is sufficient to bring out the requisite hardness in one, will be found quite inadequate to produce the same effect in another; the only certain mode of proceeding is to take a portion of the bar which is intended to be worked up into tools, commencing first *below* the presumed hardening heat, and if on quenching it should prove soft, heating it again a little *higher,* and so on by degrees until the proper, that is, the *lowest degree* of heat at which it will harden, is ascertained; this once known should be carefully observed, and all articles made from the same bar heated to that limit and no higher.—*Gill's Technical Repository,* vol. i. p. 138, 139.

It is a favourite theory with some, that the different degrees of heat suitable for various tools, may be ascertained with sufficient accuracy for all practical purposes without having recourse to the ulterior process of tempering; or in other words, that by heating pieces of steel to different degrees before plunging them into the water, the one heat attains full hardness, the next the temper of a tool fit for metal, another of a tool fit for wood, a fourth that of a spring, and so on. This opinion is in some measure corroborated by the fact that, if the end of a piece of steel be made fully hard, the transition is nearly, but not quite immediately from the hard to the soft part; and Mr. Ross, in making the cutters or dividing points for his dividing engine, hardens the end of a longer piece of steel than is required, and forms the point upon the grindstone, exactly at that part where the temper suits without the steel being let down at all, a practice originally introduced by Mr. Stancliffe, a workman of the celebrated Ramsden. His method was as follows: After shaping the tool and condensing it by hammering, he carefully heated the point and quenched it; then with the edge of a file he made trial by filing along from the soft and unhardened part, to that part where it became hard, and formed the cutting edge by grinding and whetting *that* part to shape. He was thus assured of the quality of his tool being the best that the steel employed could possibly produce, nor did it require tempering in the usual manner.

To the working mechanic this mode of hardening without tempering is rendered unavailable, except in some few particular instances. The time employed and the skill required are serious drawbacks to its adoption; and after all, the scale is confined within such extremely narrow limits as to be, generally speaking, practically valueless.

Some tools require to be tempered after being hardened; others do not. In the first class may be included all tools which are liable to be exposed to violent strains or blows, as screw-taps, cold chisels, hatchets, caution with regard to the heat, always remembering that steel burned or overheated is not merely superficially injured, by having its surface covered with scales, but the pores of the metal become open and expanded, the firmness of its texture is destroyed, and in this state it is quite incapable of sustaining a cutting edge.*

Secondly. In order to complete the process of hardening, it is necessary to employ some cooling medium by which the heat may be rapidly extracted. Currents of cold air, flat metallic surfaces, immersion in water in various states, in oil or wax, freezing mixtures, and mercury, have been severally resorted to, all of which have proved more or less efficacious, in proportion to the good conducting powers they respectively possess. Mr. Stodart recommends simple water at a temperature of 40° Fahrenheit. There seems, however, to be an opinion, very generally entertained, that mercury imparts the greatest degree of hardness, then cold water impregnated with salt, or water medicated with divers *astringent* and *acidifying* matters. Simple water follows, and lastly oily mixtures. The high place mercury occupies in the scale, may be attributed to the superior conducting power of the metal, and the non-production of steam, which is necessarily generated when water is used, and, as some think, causes a separation between the steel and the water. It is difficult to assign any satisfactory reason for the advantage of salt and water, unless, indeed, the fluid thus treated derives from its greater density a superior conducting power. It has, however, this disadvantage—all articles hardened in it have a tendency to rust, and this can only be corrected by the use of lime-water, or some other neutralizing agent. The file-makers adopt a method peculiar to themselves; but as the secret has never been wholly divulged, our information on this head rests mainly on supposition. The cooling medium is stated to be water saturated with salt, to which a little white arsenic is added; and it has been suggested that a small proportion of the arsenic enters into combination with the steel, and thus contributes an extraordinary degree of hardness.†

Many workmen are favourably inclined to water that has been in constant use during a number of years, provided it be free from grease. When the steel is harsh, it is advisable to take the chill off plain water, and this precaution is thought to lessen the risk of cracking. Oily mixtures impart to thin articles, such as springs, &c., a reduced yet sufficient degree of hardness, and certainly with less risk of cracking than from water; in some cases it is found expedient to cover the water with a thin film of oil, which is considered to assuage, and in some measure counteract the sudden transition from the extreme of heat to that of cold.

What is designated, though not very appropriately, a *natural* spring, is made with a vessel having a true and false bottom, the latter perforated with small holes; it is filled with water, and a copious supply of the fluid admitted beneath the partition, it ascends through the holes and pursues the same current as the heated portions, which also escape at the top. This contrivance was invented by the late Mr. Oldham, engineer to the Bank of England, and was used by him in hardening the rollers for transferring the impressions to the steel plates for bank notes.‡

A somewhat analogous process was adopted by Mr. Adam Eckfeldt of the United States Mint in hardening steel dies. He caused a vessel, containing 200 gallons of water, to be placed in the upper part of the building, at the height of forty feet above the room in which the dies were hardened; from this vessel the water was conducted down through a pipe one inch and a quarter in diameter, with a cock at the bottom, fitted with nozzles of different sizes, to regulate the bulk of the jet of water which was directed to the centre of the heated die.§

Steel may be hardened as well under cover as if exposed, provided it be cooled in the requisite time. Delicate articles are occasionally inclosed in a metal box perfectly air-tight, and thus subjected both to the heat of the fire and the cold temperature of the water without bringing the articles in question into immediate contact with either. Some mechanists are accustomed to coat steel articles with a thin layer of loam mixed with water, by dipping them into the mixture and gradually drying them before putting them into the fire, in order to avoid oxidizing or scaling the steel; the loam instantly flies off the surface of the articles when plunged into water, and therefore does not in the least impede the operation. In some cases the adjoining parts of a piece of machinery are required to be respectively hard and

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&c.; in the second many tools for metal, used with quiet and uniform pressure, which are left of the full hardness for greater durability. The quality of the steel must also be taken into consideration, as cast steel let down to a straw colour is softened nearly as much as other kinds to a purple or blue. As a general rule it may be stated, that if a tool will stand *without* tempering so much the better, but the ignition must be carefully regulated in the first instance, as the most useful hardness is produced by that degree of heat which is just sufficient to effect the purpose, and it is clear that if the precise limit be once passed, we must then have recourse to tempering.

[*] A very general but erroneous opinion is entertained, that if steel be overheated previously to immersion, an extra amount of heat is likewise required to reduce it, or what is termed *let it down,* to a proper degree of hardness, and that, without this, a good cutting edge cannot be produced. This, to say the least of it, is a miserable and inefficient attempt to remedy one error by the introduction of another. It may be laid down as a position which is not in much danger of being controverted, that the *lowest possible heat* at which steel can be hardened is unquestionably the best; and that no degree of *temper* whatever will operate to restore to steel the properties of which it has been deprived by being *overheated.*

[†] The celebrated files made by Raoule of Paris, are stated to be so extremely hard that they will act upon or abrade most files of English manufacture; this peculiarity is supposed to be given by the following compostion:—

Mutton suet, *not rendered,* but chopped small, .	2 lbs.
Hog's lard, ...	2 lbs.
White arsenic in powder, ...	2 oz.

These ingredients being put into an iron vessel, with a cover fitted to it, must be boiled until all moisture is driven off. It is advisable to perform this operation, as well as that of quenching any article in it, under the hood of a smith's forge, in order to avoid inhaling the noxious arsenical fumes which are disengaged. In using this compound it must be first melted, and any thin article may be placed upon a red-hot plate of iron to receive the proper heat, and then immersed in the liquor.—*Gill's Technical Repository*, vol. i. p. 212, 213.

[‡] Holtzapffel's Mechanical Manipulation, vol. i. p. 243.

[§] Gill's Technical Repository, vol. x. p. 359—361.

soft. The slow conducting power of lamp-black has been found a very useful agent in protecting any particular part, during the process of hardening the remaining portion. For example, suppose we wish to harden the neck of a lathe-mandrel, but not the outside screw, lest it should break when roughly used; this may be done by means of an iron tube, fitted a little way on the neck of the mandrel, and ramming the intervening space between the screw and the interior surface of the tube full of lamp-black, so as completely to envelop the screw; the orifice of the iron tube or case must then be closed with a metal plug; the mandrel being hardened in the usual way, the exposed parts will be hard, and the protected portions will remain soft.

The process of hardening is invariably attended with anxiety, and frequently uncertain in its effects, arising principally from the sudden change from heat to cold, which often causes the work to twist, or become otherwise distorted, if not absolutely cracked. The danger of the last accident is much increased with thick massive pieces, which appear to be hardened in layers, or, at all events, the outer crust may be perfectly hard, while the inner portions are less and less hard as they approach the centre, which, in many instances, is found to be so soft as to yield easily to the file. The most rational explanation of this fact is, that steel under the influence of heat becomes expanded, but when plunged in water, the outer crust is made to contract from the abstraction of heat; but this loss of heat goes on less rapidly at the centre than at the exterior parts, whence it may be assumed, that the inner bulk continues to contract after the outer case is fixed, and thus tends to tear the two asunder, the more readily if there happen to be any flaw or defect in the steel itself. Nor is it by any means unusual to find an external flake shell off in hardening; and it sometimes happens that work which is apparently sound when removed from the water will eventually give way, perhaps after the lapse of some hours, to the unequal tension produced by the violence of the hardening process, and crack with a loud report.

As regards both cracks and distortions, we have the authority of a competent judge of great practical experience for stating that, generally speaking, their avoidance depends principally upon *manipulation or the successful management of each step;* first, the original manufacture of the steel, its being forged and wrought so that it may be equally condensed on all sides with the hammer, otherwise, when the cohesion of the mass is lessened, from its becoming red-hot, it recovers in part from any unequal state of density in which it may have been placed.*

It must be remembered also that steel at a red heat is in its weakest condition, and, consequently, liable to injury from two causes. First, from careless handling with the tongs; and, secondly, from encountering the sudden cooling action irregularly; it is therefore considered the safest practice, to plunge works vertically, as all parts are then similarly exposed to equal circumstances, and less disturbance is risked than when the objects are immersed obliquely or sideways into the water.†

In all cases the thick unequal scale left from the forge should be removed. The vicious and most defective process of hardening direct from the anvil, cannot be too strongly deprecated, since it must, we apprehend, be sufficiently obvious to any person, who will take the trouble to disabuse his mind of an old custom which has nothing but antiquity to recommend it, that steel hardened with the scale adherent to its surface must naturally receive a very imperfect impression either of heat or cold, because this scale, which is produced by the act of forging, varies in thickness according to the degree of heat employed in forging; it is also a bad conductor of heat; consequently, the sudden transition from heat to cold, on which the success of the operation depends, is not uniform in respect of time, the results of which are shown by one piece of steel being hard, while another is scarcely hardened at all, owing to the resistance presented by the scale to the cold temperature of the water. Hence, when such articles are *drawn down*, scarcely any two out of a large number will be found alike in temper, a fact which can scarcely have escaped the observation of any one in the habit of using edge-tools.

As a means of avoiding this slovenly and very imperfect mode of hardening, all that is requisite is to pass the material from the anvil to the grindstone, a slight application of which will remove the whole of the scale or coating, and render the steel fit for the operation of hardening with that accuracy which cannot otherwise be obtained; for it is undoubted that steel, thus cleansed, heats in the fire with great regularity, and is equally sensible to the immediate action of the water on its body when immersed, so as to receive a similar degree of hardness from one extremity to the other.

Thirdly, with regard to the heat for tempering or letting down.

It is a singular fact that steel should be softened by the same means that are ordinarily employed to harden it. This tempering or reduced hardness is given by exposing the article to a certain temperature, the choice of which appears to be rather a matter of experience than deduced from any well-defined principle, since the problem has never yet been satisfactorily solved, why a particular temperature imparts a particular temper; consequently, each workman uses such a heat, and tests it by such signs as seem best to accord with the result of his own experience. Two modes of tempering hardened steel are in common practice; the first is termed *blazing:* the article is smeared with tallow or oil, and held over the fire till the smoke rises copiously; this is supposed to indicate a temper equivalent to 450° Fahrenheit; a still lower temper is given when the oil or tallow catches fire, and from this the process is said to derive its distinctive appellation "blazing off." This method of lowering or drawing down is manifestly very indefinite and defective, as it affords no accurate means of estimating the effect produced by certain degrees of heat on hard steel of different and varying quality, yet it is very extensively applied, and in some branches of hardware manufacture to the exclusion of every other. A convenient mode of tempering any number of articles at once, and of heating them uniformly, however irregular their shape, is to put them into a proper vessel, with as much oil or tallow as will cover them; the vessel is then placed on the fire, and the temper determined by the following indications:—When the tallow is first observed to smoke, it denotes the same temper as that called a straw colour; when the smoke becomes more abundant, and of a darker colour, it will be equal to a golden brown: if the tallow be heated so as to yield black smoke, and in a plentiful quantity, this will give a temper equivalent to purple. The next degree of heat is known by the tallow taking fire, if a lighted body be presented to it, yet not so hot as to continue to burn when the light is withdrawn; this indicates a blue. If the whole of the tallow be allowed to burn away, or burn dry, to use the technical expression, it imparts the temper generally selected by clockmakers for their work. A further supply of tallow would be useless, as a very trifling degree of heat, beyond this point, would just be visible in a dark place, which is equivalent to the lowest grade of red heat.

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* Holtzapffel's Mechanical Manipulation, vol. i. p. 244.

† Some articles require a different treatment, but as they do not come within the scope of the present inquiry, we merely remark that for swords and objects of similar form, it is found best to dip them exactly as in making a vertical downward cut with a sabre, which for this weapon is its strongest direction.

The second method has reference to the colour or film of oxide* developed on the bright surface of hardened steel by the application of heat; this, perhaps, for all practical purposes is the simplest and most direct criterion, as it enables the workman to form a tolerably correct judgment of the intermediate grades between the two extremes of hard and soft steel. If, however, we wish to arrive at extreme accuracy, we must resort to the use of a bath and thermometer; the bath may be of mercury, or of the fusible mixture of lead, tin, and bismuth, or, indeed, of any fluid, whose boiling point is not much under 600°. Into this bath the articles to be tempered are put together, with the bulb of a thermometer graduated to near the boiling point of mercury. Some years ago Mr. Stodart instituted a series of experiments, in order to determine the temperatures, at which the several colours make their appearance upon hardened steel; the results are shown in the following tabular arrangement:—

1.	Very pale straw yellow,	430°	Fahr.	Tools for metal.
2.	A shade of darker yellow,	450°	„	
3.	Darker straw yellow,	470°	„	Tools for wood, screw-taps, &c.
4.	Still darker straw yellow,	490°	„	
5.	A brown yellow, .	500°	„	Hatchets, chipping chisels, and other percussive tools, saws, &c.
6.	A yellow, slightly tinged with purple, .	520°	„	
7.	Light purple, . .	530°	„	
8.	Dark purple, .	550°	„	Springs.
9.	Dark blue, . .	570°	„	
10.	Paler blue, .	590°	„	Too soft for the above purposes.
11.	Still paler blue, .	610°	„	
12.	Still paler blue, with a tinge of green,	630°	„	

The first change or tint is at about 430°, but this is too faint to be distinguished except by comparison with another piece of untempered polished steel.

The degrees at which the respective colours are produced being thus known,† it follows, that the workman has only to heat the bath with its contents up to the required point. The advantages attending this method are obvious; the heat is equally applied, and the workman, instead of attending to the colour, has merely to observe the thermometer.‡

Antecedent, however, in point of time to the process just described, is that introduced by Mr. David Hartley, and for which, we believe, he obtained letters patent. Mr. Hartley's plan was to immerse the hardened steel articles in a bath of oil heated to a regulated temperature, and measured by a thermometer. This was unquestionably the first decided step towards improvement, both as regards precision and despatch, on the common method of ascertaining the temper by the colour.[*]

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* That the colour produced on the surface of heated steel is the effect of oxidation, is proved from the circumstance, that when steel is heated and suffered to cool under mercury or oil, none of the colours appear, nor do they when it is heated in hydrogen or nitrogen.—*Brande's Manual of Chemistry*, vol. ii. p. 42.

† The French nomenclature of colours differs in some respects from Mr. Stodart's arrangement. The following shades are specified in the *Dictionnaire Technologique*:—1st, *yellow*, which also has three distinct shades, *straw-colour*, *canary-colour*, and *lemon-yellow*; 2nd, *orange*; 3rd, *red*; 4th, *violet*; 5th, *blue*; 6th, *grey*. When steel arrives at this last colour, however little the heat may be increased, it returns to its first state, that is, as it was before being hardened and tempered.

It must be remembered too, that the tempering colours differ slightly with the various qualities of steel, and also that the intermediate shades produced on steel by heat between 500° and 580°, are frequently exhibited irregularly on different parts of the surface. "As I had," observes Mr. Nicholson, "noticed this irregularity particularly on the surface of a razor of Wootz, and had found in my own experience that the colours on different kinds of steel do not correspond with like degrees of temper, and probably of temperature, in their production I was desirous that some experiments might be made upon it by Mr. Stodart. Four beautifully polished blades were therefore exposed to heat in a bath of fusible metal. The first was taken up when it had acquired the fine yellow or uniform deep straw-colour. The second remained till the part nearest the stem had become purplish, at which period a number of small round spots of a purplish colour appeared in the clear yellow of the blade. The third was left till the thicker parts of the blade were of a deep ruddy purple, but the concave face still continued yellow. This also acquired spots like the other, and a slight cloudiness. These three blades were of cast steel; the fourth, which was made out of a piece called Styrian steel, was left in the bath till the red tinge had pervaded almost the whole of its concave face. Two or three spots appeared upon this blade, but the greater part of its surface was variegated with blue clouds, disposed in such a manner as to produce those waving lines which in Damascus steel are called the water. Two results are more immediately suggested by these facts; first, that the irregular production of a deep colour upon the surface of brightened steel, may serve to indicate a want of uniformity in its composition; and secondly, that the deep colour being observed to come on first at the thickest parts, Mr. Stodart was disposed to think that its more speedy appearance was owing to those parts not having been hardened."—*Martin's Circle of the Mechanical Arts*, p. 262.

‡ Some recent experiments proving that steel, for certain uses, is sufficiently tempered long before it is heated to produce any change of colour, promise to give additional value to this process by a thermometer. The knife edges attached to the pendulum described by Captain Kater (Phil. Trans. 1818, p. 38), were forged by Mr. Stodart from a piece of fine wootz. They were carefully hardened and tempered in the bath at 430°; on trial they were found too soft. They were a second time hardened and heated to 212°. The intention was to increase the heat from that point, trying the temper at the advance of about every ten degrees. In the present instance this was not necessary, the heat of boiling water proving to be the exact point at which the knife edges were admirably tempered.

It is highly probable that steel, for many uses, may be sufficiently tempered in a range so extensive as from 212° to 430°, and by the thermometer all the intervening degrees may certainly be ascertained —*Brande's Manual*, vol. ii. p. 41, 42.

As the heat required for tempering is extremely moderate, it is frequently supplied by some part of the work which does not require to be hardened, and therefore is not quenched in the water. Hatchets, cold chisels, and many similar tools, in which the total bulk is considerable, compared with the part to be hardened, are heated in the following manner. In the first instance, as a necessary precaution, the workman ascertains, by means of a file, whether the article is sufficiently hard; he then as quickly as possible brightens some portions of the surface with a piece of grindstone or an emery stick, to enable him to see the colour, which attained, the work is cooled in the usual manner, lest the body of the tool should continue to supply heat. But should the desired colour not appear, recourse must then be had to some method of increasing the heat, such as the surface of a clear cinder fire, taking care to direct the heat rather above than immediately upon the part where the colour is to be observed. Another and very general manner of tempering articles of moderate size is to heat to redness a portion of one end of a flat bar of iron; the tools being previously brightened, are then placed upon that part of the heated bar which is found, upon trial, to impart the necessary degree of heat, by gradually moving the work to and from the heated extremity, and as the required colour becomes apparent, it is pushed off the bar into a vessel of oil or water. But this process is only applicable when the articles contain a small body of metal pretty equally distributed, and when some oxidation of the surfaces is not seriously objectionable.

Some articles may with advantage be hardened and tempered at one operation. Springs especially, if made of steel wire or sheet steel, subjected to the treatment we are about to describe, are found to possess much more elasticity, and are less liable to crack in hardening than by the usual process.

The springs are raised to the proper degree of heat by being placed in a crucible in a suitable furnace; but instead of being quenched in oil or water, they are plunged into a metallic bath, kept over another furnace at a heat a little *below* the tempering point, which is ascertained by means of a pyrometer, immersed in the bath, and thus they are not cooled *entirely*, as in the ordinary methods, which renders them exceedingly liable to crack in hardening, but only to *the proper tempering degree.* The metal for the bath may be an alloy of tin and lead, which is capable of fusing, at or just below the requisite temperature. The pyrometer consists of a slip of brass and one of steel riveted together, which are firmly secured at one end to a metal plate, forming the base of the instrument, and at the other act on the shorter arm of a lever or index turning on a pivot or centre, and whose longer arm indicates the degree of heat on a graduated arc formed upon the surface of the metal plate. The instrument should be inclosed in a metal case, to protect it from becoming clogged by contact with the fused metal. The heat of the bath may be lowered from time to time, as required, either by adding more metal, or by abating the heat of the furnace.

The springs are finally quenched in oil, and then blazed off, which not only imparts additional toughness, but coats the exterior with a sort of varnish, and thereby acts as a preservative against rust.

This method is similar in principle to that adopted by Mr. Perkins, who discovered that steel blocks did not warp or crack in hardening, when cooled down *nearly* but not *entirely*, as is usually done, to the tempering heat.

Oil, or various combinations of oil, tallow, resin, wax, &c. are respectively used for many thin and elastic articles, such as saws and springs, which require a lower or milder degree of hardness than is imparted by water. Mr. Holtzapffel[†] mentions that the composition used by an experienced saw-maker is 2lbs. of suet and ¼lb. of bees-wax to each gallon of whale oil; these are boiled together, and will serve for thin works and most kinds of steel. The addition of black resin, to the extent of about 1lb. to the gallon, makes it serve for thicker pieces, and for those it refused to harden before; but the resin should be added with judgment, else the works will be rendered too hard and brittle. The composition is useless when it has been constantly employed for about a month; the period depends, however, on the extent to which it is used, and the trough should be thoroughly cleaned out before new mixture is placed in it.

The following recipe is highly recommended by Mr. Gill.[‡]

Spermaceti oil, ...	20 gallons.
Beef suet, *rendered*, ...	20 lbs.
Neat's foot oil, ...	1 gallon.
Pitch, ...	1 lb.
Black resin, ...	3 lbs.

These last two articles must be previously melted together, and then added to the other ingredients, when the whole must be heated in a proper iron vessel with a close cover fitted to it until all moisture be evaporated, and the composition will take fire on a flaming body being presented to its surface; this state once ascertained, the lid or cover must be immediately replaced and the flame extinguished.

Cast steel articles, if *thin* or *slender*, may be quenched in this composition in order to harden them, and then be blazed off over a clear fire in order to temper them; but if the articles are *thick*, such as sword blades, &c., they should previously be hardened by immersion in rain water, and then prepared for tempering by applying a thin coating of the liquid, by means of a hard brush, on both sides, and blazed off to the proper temper.

The next example we shall take, is that of Mr. Adams' patent bow-springs for carriages, some of which measure 3½ feet long, and weigh 50 lbs. each piece, two of them being used in combination; other single springs are 6 feet long, and weigh 70lbs.

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[*] It is evident that a bath of any of the soft metals, whose fusible points are just above that required for tempering, may be used instead of oil, and alloys of those metals might be so proportioned as to obtain points of fusion at the exact degree of heat required. In these cases, however, to guard against oxidation, it would be advisable to keep the fluid metal covered with grease. Oil is preferable to the fusible mixture for several reasons. It is cheaper, and admits of the work being seen during the immersion, by reason of its transparency.

[†] Mechanical Manipulation, vol. i. p. 269.

[‡] Technical Repository, vol. i. p. 257, 258.

In hardening them, they are heated by being drawn backwards and forwards through an ordinary forge-fire, built hollow, and they are immersed in a trough of plain water; in tempering them, they are heated until the black red is just visible at night, by daylight the heat is denoted by its making a piece of pine wood sparkle when rubbed on the spring, which is then allowed to cool in the air. The metal is $\frac{9}{16}$ths of an inch thick, and Mr. Adams considers $\frac{5}{8}$ths the limit to which steel will harden properly, that is, with sufficient regularity to serve as a spring.

In hardening steel, more particularly in massive blocks, or those which are unequally thick and thin, there is no greater or more perplexing difficulty than their tendency to crack from the sudden change from heat to cold. And in reference to hardening razors, an analogous case, Mr. Stodart records, as the observation and practice of one of his workmen, that the charcoal fire should be made up with shavings of leather, and upon being asked what good he supposed the leather could do, this workman replied, "that he could take upon himself to say that he had never had a razor crack in the hardening, since he had used this method, though it was a frequent occurrence before."

"When," observes Mr. Stodart, "brittle substances crack in cooling, it always happens from the outside contracting and becoming too small to contain the interior parts. But it is known that hard steel occupies more space than when soft, and it may easily be inferred that the nearer the steel approaches to the state of iron, the less will be this increase of dimensions. If, then, we suppose a razor or any other piece of steel to be heated in an open fire, with a current of air passing through it, the external part will, by the loss of carbon, become less steely than before; and when the whole piece comes to be hardened the inside will be too large for the external part, which will probably crack. But if the piece of steel be wrapped up in the cementing mixture, or if the fire itself contain animal coal, and is put together so as to operate in the manner of that mixture, the external part, instead of being degraded by this heat, will be more carbonized than the internal part, in consequence of which it will be so far from splitting or bursting during its cooling, that it will be acted upon in a contrary direction, tending to render it more dense and solid. The cracking which so often occurs on the immersion of steel articles in water, does not appear to arise so much from any decarbonization of the surface merely, as from the sudden condensation and contraction of a superficial portion of the metal, while the mass inside remains swelled by the heat, and probably expands for a moment on the outside coming in contact with the water."

The file makers adopt a precaution peculiar to themselves, which is considered to obviate in a great degree the risk of *clinking*, or partial cracking in the hardening process. The files are drawn through yeast or beer grounds, and then through a mixture of common salt and animal charcoal. This treatment is plainly corroborative of Mr. Stodart's opinion, inasmuch as it contributes a small extra dose of carbonaceous matter to the outside, and also renders the steel rather harder and less disposed to crack;* the composition has also the advantage of protecting the fine points of the teeth from being injured in the fire.

In hardening engraved plates, rollers, dies, and articles of similar character, it is manifestly of the last importance to preserve the surface clean and unimpaired, because steel at a red heat is extremely liable to oxidation if exposed to the atmosphere even for a few seconds, and this oxidized scale will, in many cases, nearly obliterate, or, at all events, seriously injure the engraved surface; it is, therefore, a matter of the utmost consequence to conduct the heating and cooling with the most complete exclusion of the air. Mr. Thomas Oldham has lately introduced a mode of proceeding, which, in the opinion of competent judges, is thought to approach as nearly as possible to perfection. By it the work, instead of acquiring the ordinary black and gray tints, and a minute roughness like the surface of the finest emery paper, is made to come out of the water as smooth to the touch as at first, and mottled with all the beautiful tints seen on case-hardened gun-locks; the method is simply as follows:—

The work to be hardened is inclosed in a wrought-iron box with a loose cover, a false bottom, and with three ears projecting from its surface about midway; the steel is surrounded on all sides with carbon from leather, driven in hard, and the cover and bottom are carefully luted with moist clay. Thus prepared the case is placed in a vertical position, in a bridle fixed across a great tub, which is then filled with water almost to touch the false bottom of the case. The latter is now heated in the furnace as quickly as will allow the uniform penetration of the heat. When sufficiently hot, it is removed to its place in the hardening tub —the cover of the iron-box is removed, and the neck, or gudgeon of the cylinder, is grasped *beneath the surface of the carbon*, with a long pair of tongs, upon which a coupler is dropped to secure the grasp. It only remains for the workman to hold the tongs with a glove whilst a smart tap of a hammer is given on their extremity; this knocks out the false bottom of the case, and the cylinder and tongs are instantly immersed in the water; the tongs prevent the cylinder from falling on its side, and thus injuring its delicate, but still hot surface. For square plates a suitable frame is attached by four slight claws, and it is the frame which is seized by the tongs; the latter are sometimes held by a chain which removes the risk of accident to the individual. In some cases the work assumes a striated and mackled appearance, evident to the touch as well as the sight, and which is to be attributed to an imperfect manufacture of the steel.† Various attempts have likewise been made to prevent the warping or other distortion to which steel is greatly disposed in the operation of hardening. Mr. Rehé was accustomed to harden small cylindrical wires, previously heated to the proper degree, by placing them upon the flat surface of a fixed block of cast-iron, and giving them a rolling motion by sliding another flat plate of iron over them, by which means he found them set perfectly straight.* "This process," observes Mr. Holtzapffel, "might probably answer, but unfortunately cylindrical steel wires supply but a very insignificant portion of our wants."

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* Occasionally the prussiate of potash is applied to the surface of steel, in order to impart a greater amount of carbon, and consequently render it harder, as it is also used to convert iron superficially into steel, or case-harden it.

† Mechanical Manipulation, vol. i. p. 255, 256.

Another mode suggested by Dr. Wollaston was to enclose the piece of steel in a tube or case surrounded with Newton's fusible alloy, the whole to be heated to redness and plunged into cold water; the work was readily released by immersion in boiling water, which melted the alloy, and the article came out hard and quite perfect in form.

Mr. Perkins found that by boiling the water, in order to expel the air, in which his engraved steel-plates were to be hardened, and plunging them vertically the risk of distortion was much reduced. Long experience shows, that on plunging the heated steel into water, and allowing it to remain there no longer than is sufficient for lowering the temperature of the steel to the same degree as that to which a hard piece of steel must have been raised, in order to temper it in the common way, not only produces the same degree of hardness in the steel, but what is of much more importance, almost entirely does away the risk or liability of its cracking or breaking. It is impossible to communicate by words, or to describe the criterion by which we can judge of, or determine when the steel has arrived at the proper degree of temperature after being plunged into cold water; it can only be learned by actual observation, as the workman must be guided by the kind of hissing or singing noise, which the heated steel produces in the water while cooling. From the moment of its being first plunged into the water a varying sound will be observed; and it is at a certain tone before the noise ceases that the effect to be produced is known. The only directions which can be given whereby the experimentalist can be benefited are as follows: namely, to take a piece of steel which has already been hardened by remaining in the water till quite cold; and by the common method of again heating it, to let it be brought to the pale yellow or straw colour, which indicates the desired temper of the steel plate to be hardened by the above process; so soon as he discovers this colour to be produced, to dip the steel into water, and attend carefully to the hissing noise which is produced; he will then be better able, and with fewer experiments, to judge of the precise time at which the steel should be taken out. It is not meant to be understood that the temper indicated by a straw colour is that to which the steel plate, cylinder, or die, should be ultimately reduced; because it would then be found too hard; but merely that the temperature which would produce that colour, is that by which the peculiar sound would be occasioned when the steel should be withdrawn from the water for the first time. Immediately on withdrawing it from the water, the steel plate, cylinder or die, must be laid upon, or held over a fire, and heated uniformly, until its temperature is raised to that degree at which tallow would be decomposed; or, in other words, until smoke is perceived to arise from the surface of the article after having been rubbed with tallow. It must then be again plunged into water, and kept there until the sound becomes somewhat weaker than before. It is then to be taken out and heated a second time to the same degree, by the same test of smoking tallow as before, and the third time plunged into water till the sound becomes again weaker than the last. Expose it a third time to the fire as before, and for the last time, return it into the water, and cool it; after it is cooled clean the surface, and by heating the article over the fire, the temper must be finally reduced by bringing on a brown, or such other lighter or darker shade of colour, as may best suit the quality of the steel, or the purpose to which it is to be applied.†

The foregoing observations seem to justify the conclusion that it is the best and safest practice, in regard to articles of much substance, to combine the hardening and tempering processes, instead of allowing the work to become entirely cold, and then to re-heat it for tempering. A brief summary of the more important points relative to the subject in hand, and upon which all are agreed, may be desirable; this may be expressed in very few words.

The degree of heat suitable to forging and hardening steel varies in some measure with its quality and mode of manufacture; the heat is in an inverse ratio to the increase of carbon.

In every case, *the lowest possible heat*, at which steel can be worked and hardened, is undoubtedly the best—the hammering should be applied in *the most regular and equal manner throughout;* and for cutting tools it should be continued till they are nearly cold.

Coke, cinders, or charcoal, are decidedly preferable as fuel to fresh coal, the sulphur of which is most detrimental.

The scale adhering to the surface of the work should be invariably removed, so as to expose it more uniformly to the cooling medium. Hardening a second time, without having recourse to hammering, is attended with considerable risk, and the less frequently steel is subjected to the action of the fire the better.

In conclusion, the workman will do well to remember, that he who desires to earn a name for his work, should assiduously and patiently study the quality of steel, in order to treat it accordingly, and, moreover, diligently note those apparently trivial indications, which escape the eye of an unpractised observer; but above all he must disabuse his mind of the common prejudice, that no one can teach him anything concerning steel.

In accordance with the arrangement mentioned at the commencement of this article (p. 154), we have now to consider tools which receive motion from the hand of the workman. Of these our notice must be brief, as their number is considerable, and our attention being directed to the working of metals, we shall necessarily confine our observations to such tools as are used in that branch of operative mechanics.

The metals are commonly prepared for ulterior operations by *chipping*, which not only produces the intended effect in an expeditious manner, but also saves the expense of files, which otherwise would be required. This process is most usually applied to cast iron, the rind, or exterior coat of which, taken as it comes from the mould, is invariably much harder than the interior, and frequently so extremely hard that it would spoil the best file in a few minutes, whilst the interior immediately beneath the scale is found to be nearly as soft as brass.*

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* Gill's Technical Repository, vol. ii. p. 193.

† Trans. Soc. of Arts, vol. xxxviii. p. 54, 56.

The tool employed for chipping is simply a chisel with an edge assuming the shape of an acute wedge; it is ordinarily made from square or oval steel of the best quality, rather spread out at that end which is intended to form the edge, so as to afford a greater surface. Whatever may be the length of the chisel, whether *six* or *eight* inches—and this must depend in some measure upon the nature of the work—the form of the cutting edge is in all cases nearly similar; observing, however, that it is advisable to have the chisel drawn out by the smith, by which precaution the edge, when injured, may be more easily restored on the grindstone. The operation of chipping is materially facilitated by the use of the *cross-cutting* chisel, of which fig. 7 shows a front, and fig. 8 a side view, *aa′*, in the latter figure being a section. The cutting edge of this extremely useful tool varies in breadth from one-sixteenth to five-sixteenths of an inch; its utility and application will probably be rendered more obvious to the reader by a diagram, than by any lengthened verbal explanation.

Fig. 7.

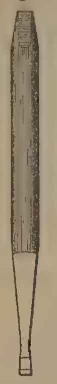
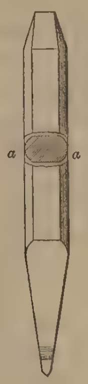


Fig. 8.



Suppose the surface of a block of cast iron, represented by fig. 9, to require chipping. In the first place, the workman cuts longitudinal grooves *aa*, *bb′*, *cc′*, throughout or across the entire length or breadth of the surface by means of the cross-cutting chisel, and at such a distance from each other as is rather less than the width of the chipping chisel intended to be subsequently employed; by which means the corners of the edge of the chipping chisel are essentially preserved from injury, as under ordinary circumstances it is found that the corners of the chisel first give way, and require constant repair.

Fig. 9.

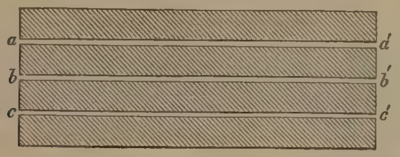


Fig. 10.

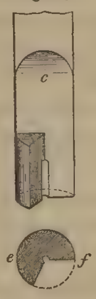


Fig. 11.

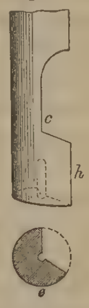
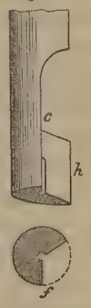


Fig. 12.



Cylindrical holes are generally pierced in any piece of metal by a tool called a drill. Boring differs from drilling principally, as we shall hereafter show, in being applied to larger works. The class of tools which come within the general description of drills, or cutters, is extremely numerous; that more commonly employed is too well known to require description, more especially as we have already illustrated an improved form of this tool, (p. 159, fig. 6.) The pin drill, and half-round drill are, in certain cases, extremely useful; the only objection to the former is, that it requires a small hole to be first cut in and through the metal in which the pin of the drill works and necessarily follows; it answers, however, for all ordinary purposes, and performs its work extremely well, although it cannot be depended on in cases where rigid accuracy is required. The half-round drill offers little or no security whatever, as regards piercing in a right line; it is, however, a very useful tool, and may, in many instances, be advantageously employed. Perhaps the most effective form of drill yet introduced, more especially if applied to any metallic substance revolving in a lathe, is that invented by M. Collas, an eminent French mechanician. This tool (of which fig. 10 is a front view, and figs. 11, 12, side views, taken from *e*, and *f*), is turned truly cylindrical throughout its entire length, except at that end which is intended to fit into the brace, or if used in a lathe a small portion of the metal is filed square, or flattened in order to admit of any convenient mode of preventing the drill turning. At the other or opposite extremity of this cylinder of steel, and through the centre, a hole is drilled, proportionate in size to the size of the tool; one half of a portion of the bar is then cut away, leaving the remainder cylindrical; this part is then equally divided into three, and one of them filed out, as is clearly shown in the plan views of figs. 10, 11, 12, by which process the central hole is cut into two equal parts, and becomes a small semicircular groove. With regard to the angle of inclination to be given to the cutting end of the drill, this must depend principally upon the resistance offered by the material.† This tool manifestly cuts circularly, except at the centre, where it forms a small projecting pin, which enters the central groove and serves as a conductor; in proportion as the tool advances this pin increases in length until it reaches the extremity of the groove, when it necessarily breaks and comes away with the chips. It is important to observe that this groove must be rather less than a semicircle, otherwise the pin of metal which enters it, being cylindrical, would tend to leave it during the progress of the operation, and the distinguishing feature of this tool would be destroyed.

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* The crust of the casting which always retains some sand, is very destructive to the tools unless they can be sent deep enough to penetrate to the clean metal beneath; but when little is to be removed from the casting, or that they are wrought with expensive tools, circular cutters, &c., it is desirable to *pickle* the work, or, in other words, to undermine the sand by dissolving a little of the metal with acid. Iron castings are pickled with sulphuric acid diluted with about twice as much water; the castings, if small, are immersed in a trough lined with lead, or else the acid is sprinkled over them; in two or three days a thin crust, like an efflorescence, may be then washed off with the aid of water and slight friction.

Brass and gun-metal, when pickled, require nitric acid diluted with four to six times as much water, otherwise the rough coat should be removed with an old file or a triangular scraper, but which is less effective than the dilute acid; this should be also kept in leaden vessels, or in those of well-glazed earthenware, or glass.—*Holtzapffel's Mechanical Manipulation*, vol. i. p. 375.

† Drills or boring bits ought to have the angles of their edges varied according to the nature of the metal to be bored; thus, wrought iron would require a very different angle from that used for cast iron. If in use the bit trembles or jars, it is a sign that the angle is too acute, and must be made more obtuse, or nearer to a right angle with the plane or flat face of the drill. Again, if the obliquity of the other, or crossing angle, be too great, the tool will also have too great a tendency to form a nipple or cone, in the centre of the bottom of the hole, and to bore the hole gradually wider and wider, instead of truly cylindrical, as it will do when properly formed; and that fault must therefore be corrected by grinding the drill or bit so as to reduce its obliquity, or bring it nearer to a right angle with the sides of the bit.—*Gill's Technical Repository*, vol. viii. p. 231.

Small drills are commonly made of a single piece of steel wire, upon which, near to the middle, a pulley or drill-barrel is fixed. Occasionally a small mandrel is used, provided at one end with a square hole about half an inch deep, into which drills of various dimensions can be inserted. The disadvantage of this mode of construction is that the drill is rarely placed true in the mandrel, which necessarily causes it to perform indifferently; it is, therefore, but seldom employed by practical men who have the convenience of readily supplying themselves with drills of various dimensions as required.

When small drills are used they are held horizontally and kept up to the work by a breast-piece, which is usually made of wood, armed with a plate of steel superficially pierced with centre-points of different sizes, in one of which the blunt end of the drill works. The drill receives a reciprocating motion from an elastic bow, the string of which is coiled once round the pulley. Common bows are ordinarily made of stout cane, those of a better description of steel, and the string of catgut, but the strength of both must necessarily be proportioned to the size of the drill.

In order to cut large holes more force is obviously required than can be imparted by the method just described, instead of which a brace, not very dissimilar to that used by carpenters, is employed, and the drill itself is fitted as a boring bit; but with this difference in the mechanical arrangement: instead of the stock remaining stationary, we have in this case a long tapering spindle, which being nothing more than a continuation of the brace is necessarily carried round at the same time, and the motion becomes continuous. The upper part of this spindle works in an iron or steel plate, which is attached to the under side of the beam, called the drill beam. One end of this beam turns upon a transverse pin, between two uprights, pierced with various holes, to allow facility of fixing it by means of the pin at different elevations. The other end of the beam traverses between two uprights, and carries a heavy weight, which acting on the lever necessarily keeps the drill to its work, and the point of the drill being placed upon that part of the metal to be bored, the brace is revolved by the hand of the workman.

The shank of the drill should be accurately fitted in the brace, and the apparatus is generally so arranged that the work may be held in a strong bench-vice during the process.

The difficulty of applying a press or lever drill in confined situations appears to have been very generally felt. In Bergeron's *Manuel du Tourneur*,[*] there is a plan of a brace worked by a pair of bevil wheels, and Mr. George Rennie, in the last edition of Buchanan *on Mill-work and other Machinery*,[†] has given two views of a portable drill, invented by Messrs. Nasmyth, Gaskell & Co., of Manchester, which consists of a cast-iron frame, carrying an upright drilling spindle, the top of which is formed into a screw, so that it may be raised or depressed by a handle-wheel, while the requisite revolving motion is imparted to it by two small bevel wheels. When required to drill a hole in any piece of machinery, it is first of all set in its proper place; after this is done the handle or small fly-wheel is turned round for working the drill, and by a slow revolving motion given to the upper handle, communicating by means of the screw, the drill while working is made gradually to advance.

The contrivance we have now to describe is, we are informed,[‡] the invention of a practical mechanic, in the employment of Mr. Hague, of London. The distinguishing feature of this tool is the introduction of a ratchet-wheel and click, which obviate the necessity of turning the brace completely round, so that the effective power of the workman is constantly acting at the greatest advantage. Fig. 13 shows an elevation of the complete tool; fig. 14, a section of the same; and fig. 15, the ratchet with its appendages, and the arm separately. It is composed of three parts; the first distinctly shown in the sectional view, and distinguished by the letter *a*, which is simply an elongated nut; the second is a

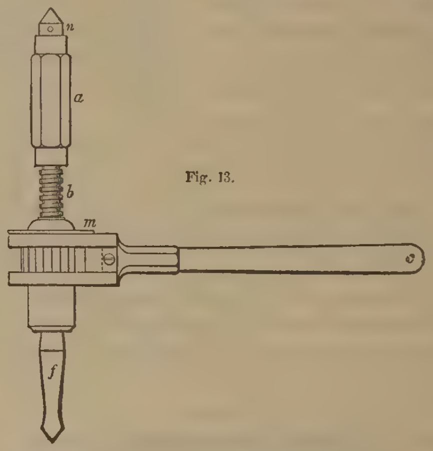

Fig. 13.

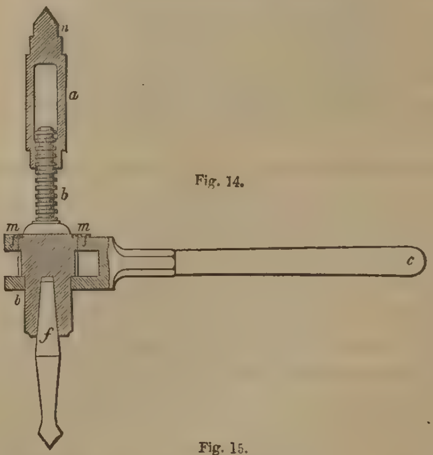

Fig. 14.

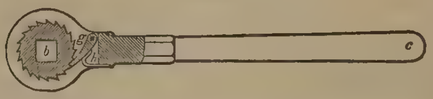

Fig. 15.

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[*] Ed. 2nde, Paris, 1816, 4to. tom. ii. pl. ix.

[†] London, 1841, royal 8vo, p. 432.

[‡] Mechanics' Magazine, London, 1835, vol. xxiii. p. 441.

cylindrical piece of wrought-iron, terminating in a square threaded screw *b*, and working into the aforesaid nut *a*. The combined length of these two, which constitute the principal part of the tool, as shown in fig. 13, may be increased and diminished at pleasure, by means of the nut *a*. The bottom piece *b* has a square hole, slightly tapered, cut in its extremity, and in a direction with its axis, for receiving the drill *f*. The handle or third portion of the tool is fitted to receive the piece *b*, being formed into a kind of fork, between the jaws of which the periphery of the piece *b* is indented so as to act as a ratchet-wheel, as may be perceived from inspection of the figures. These parts are kept in their places by the ring or cap, shown in section at *m,m*, fig. 14. The action of the ratchet-wheel is plainly restrained to one direction by means of the click *g* and spring *h*, so that when the handle is moved with a backward pull, the drill does not move.

In working, the conical point of the brace is placed under a temporary framing of iron, designated by workmen a *grab*. The tool being thus fixed is set in motion by means of the handle *c*, and the drill is pressed forward when boring, or, in technical language, "kept to its work," by unscrewing the upper portion or nut *a*, by means of a piece of pointed iron which enters a hole pierced in the solid part of the nut at *n*, for that purpose.

Such, with some unimportant modifications, is the form in which this very useful implement has till recently been constructed. Its advantages may be enumerated in few words; it requires less exertion than any form of brace or drill-stock previously introduced, and it performs its work with greater accuracy. It is not, however, free from defects: the ratchet-wheel, click and spring are liable to derangement and require to be frequently replaced, and the noise which is produced in working the instrument is far from agreeable. These objections are obviated by an arrangement recently proposed by Mr. Shanks of Johnstone, which, we think, will be found in every respect superior to the old form, and only requires to be more generally known to insure its preference and adoption. The principal peculiarity of Mr. Shanks's hand-drill is the substitution of a spiral steel band, or clutch, embracing the drill-stock and acting upon it by friction, for the more complicated combination of ratchet-wheel, click and spring. Fig. 16 shows an elevation, and fig. 17, a section of this improved form of the tool; *a* is the hollow nut for adjusting the drill to different lengths; *b*, the screw for feeding the drill; *c*, the handle of the same form, and worked in the same manner as that already described, but made with a cylindrical socket *c'*, which embraces the spiral riband *k*, and of which fig. 18 is a detached view. This spiral riband or clutch is bored truly cylindrical to fit the drill-stock *e*, and rests without being fixed upon a collar at the lower end of the stock; it is fixed to the upper part of the socket *c'* by the screw *l* and the washer *m* secures all these parts.

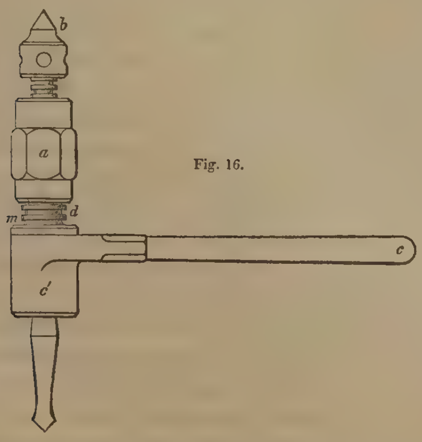


Fig. 16.

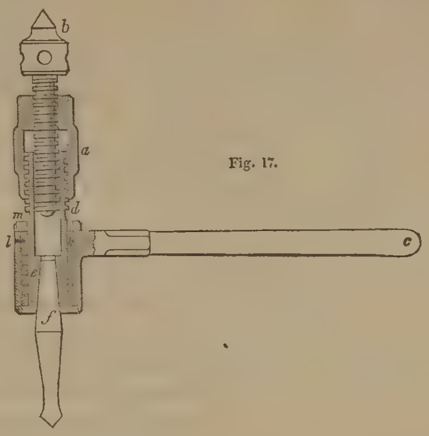


Fig. 17.

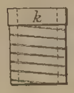
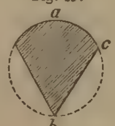


Fig. 18.

The mode in which the tool works will be sufficiently obvious from the above description, and an examination of the section fig. 17. When the handle *c* is turned in the direction in which the drill *f* cuts, the clutch *k* by its friction firmly embraces the drill-stock *e*, and turns the drill, however great the resistance may be. When the handle is returned the clutch relaxes and slips upon the stock, thereby preventing the return of the drill.

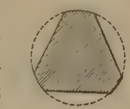
The class of tools which come within the general description of rose-bits, countersinks, wideners or broaches is far too numerous to admit of any specific description in an essay like the present. Some are intended partially to enlarge a hole previously drilled, others to do so throughout its entire depth. M. Lenseigne, a French mechanician, has made a great and decided improvement in the form of his broaches. It is a well ascertained fact that pentagonal broaches do not perform their work very accurately, more especially when applied to enlarge a hole drilled through a thin plate of metal. The motion of the brace has a tendency to render the hole sensibly larger at the mouth. To correct this defect some workmen turn the broach truly cylindrical and then remove a portion of two sides with a file, as is shown in the sectional view fig. 19: the part *a* which is a segment of a circle, bears against the sides of the hole, and serves as a conductor, and whilst the acute angular edge *b* quickly removes the material, the obtuse angle *c*, which follows, corrects any inequality of cutting. This form of broach is unquestionably preferable to any previously introduced; nevertheless it has this defect, if a chip of metal

Fig. 19.



get between the round part *a* of the broach, and the side of the hole, the angular edge *b* is necessarily thrust forward, and the truth of the work is destroyed. To obviate this objection M. Lenseigne gives to his broaches the form shown in section in fig. 20. Here there are three segments of a cylinder which serve as guides, and the metal is removed by the obtuse angular edges. A tool thus formed cuts nearly as fast and much more accurately than that shown in fig. 19; it also possesses the advantage of being more easily made than those which are either pentagonal or hexagonal.*

Fig. 20.



We have now to consider the method of cutting a screw or spiral thread upon any cylinder of metal by a manual operation. Before describing the tools by which this effect is produced, we propose to lay before the reader a brief analysis of Mr. Joseph Whitworth's excellent and thoroughly practical essay on an *Uniform System of Screw Threads,*† as applied to Bolts and Screws, used in fitting up Steam Engines and other Machinery. The difficulty of ascertaining the exact pitch of any particular thread, especially when it is not a submultiple of the common inch measure, occasions extreme embarrassment, an evil which would be completely obviated by uniformity of system, the thread becoming constant for a given diameter. The same principle would also supersede the costly variety of screwing apparatus required in many establishments, and remove the confusion and delay occasioned thereby; it would likewise prevent the waste of bolts and nuts which is at present unavoidable.

It does not appear that any combined effort has been, hitherto, made to attain so desirable an object; as yet there is no recognised standard, and this will cease to be a matter of surprise when it is considered that any standard must, to a great extent, be arbitrary. On the one hand, it is impossible to deduce a *precise* rule from mechanical principles, or from any number of experiments, and, on the other, the nature of the case is such that mere approximation would be unimportant and unsatisfactory, absolute identity of thread being indispensable. To how great an extent the choice of thread is arbitrary will appear from a cursory consideration of the principles affecting it. Without attempting to discuss these in detail, which would be foreign to the present purpose, it may be interesting to notice the general outline and bearings of the subject.

The use of the screw-bolt is to unite certain parts of machinery in close and firm contact, and it is peculiarly adapted for this purpose by the compact form in which it combines necessary strength and mechanical power. The extreme familiarity of the object tends to prevent the observation of its peculiar fitness, yet among all the applications of mechanics there is, perhaps, no instance of adaptation more remarkable. The ease with which distinct parts of machinery can be united, the firmness of the union, and the facility with which they may be separated, are conditions of the utmost importance, which by no other contrivance could be combined in an equal degree.

While, however, the utility of the screw in this application is abundantly obvious, it is by no means evident what may be the precise formation most advantageous under all circumstances. No exact data of any kind can be obtained for calculation, and the problem will be found to be capable only of approximate solution.

The principal conditions required in the screw-bolt are power, strength, and durability—the latter having reference to the wear occasioned by frequent fixing and unfixing. But none of these conditions can be reduced to any definite quantity. We cannot, for example, determine the exact amount of power necessary to draw the parts of a machine into due contact, of the precise degree of strength which may suffice for resisting the strains to which they may be exposed. Hence we cannot lay down any rule for determining the diameter of the screw-bolt required for a given purpose. Practical men can judge of the proper size with considerable accuracy, but they have no means of ascertaining it with absolute precision.

If the diameter be given, and it be required to find the proper thread, the nature of the question is not essentially altered. The amount neither of power nor of strength, nor indeed any other condition, is thereby determined. A certain limit is assigned, but within that limit the proportions of strength and power, &c., may vary indefinitely according to the actual formation of the thread. There are three essential characteristics belonging to the screw thread, namely, pitch, depth, and form. Each of these may be indefinitely modified independently of the others, and any change will more or less affect the several conditions of power, strength, and durability. The mechanical power of the screw clearly depends on the pitch, which for a given diameter determines the angle of the inclined plane, and on the form of thread which regulates the direction in which the force applied will act. The strength of the screw, as regards the thread, varies with each of the three characters; in the centre part being as the area, it is little affected, except by change of depth. The durability of the thread also depends chiefly on its depth, and the proper degree of the latter is determined principally with reference to this condition. In the selection of the thread considerable latitude of choice will be found to prevail with reference to all the characteristics; therefore no definite rule can be given for determining any one of them. It may be manifest that particular threads are too coarse or too fine, too deep or too shallow, but there are, clearly, intermediate degrees, within which the choice of thread, like that of the diameter, is arbitrary, and must be guided rather by discretion than by calculation.

The mutual dependence of the several conditions required in the thread may be noticed as having a tendency to perplex the choice. Thus, increase of power, according to a known law, is necessarily attended with diminution of strength, and the square thread which has the advantage in respect of power is proportionally weaker than the angular thread. A fine thread loses in strength, while it gains mechanically as compared with one that is coarser, and deep threads, while they are more durable than shallow, materially detract from the strength of the bolt.

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* See further Manuel du Tourneur, 2d edition, par P. Hamelin Bergeron. 4to. vol. ii. p. 36-9. Allgemeine Machinen-Encyclopädie, Leipzig, 1842. 8vo. Art. *Bohrer;* Journal des Ateliers de Tourneur, de Mécanicien, &c. par M. A. Paulin Desormeaux. Tom. i. Paris, 1830, p. 290, *et seq.*

† Communicated to the Institution of Civil Engineers, A.D. 1841.

The selection of the thread is also affected by the mutual relation subsisting between the three constituent characters of pitch, depth, and form. Each of these, as already observed, may be separately modified; but practically no one character can be determined irrespectively of the others. The pitch of the square thread is usually twice that of the angular, for the same diameter, to retain similar proportions of power and strength. Coarse threads should be deep as compared with fine to provide against the wear from friction, and a coarse angular thread will require additional depth, not only to preserve the due proportion of power, but also to prevent the longitudinal strain from being thrown too much sideways on the nut. Hence each character acts as a limit to the variation of the others; and in some instances, that is, in the case of certain diameters, it will be found that the leading considerations in fixing one character is the resulting effect on another. Thus, in some of the smaller sized screws, the pitch is determined principally by reference to the depth, a coarser thread being objectionable, because the extra depth would obviously tend to weaken the centre part of the bolt, while the necessary shallowness of a finer thread would render it too liable to wear with friction.

The proportionate strength of the thread and centre part of the screw is regulated mainly by the depth of the nut, which is usually of the same measure as the diameter of the bolt. Assuming this dimension as fixed, the proportion of strength between the two parts will necessarily vary with the different characters of thread, and more particularly with the depth. The centre part not being liable to wear, while the thread is obviously subject to friction and accidental injury, the original proportion of strength ought to be considerably in favour of the latter.

Such being the variety and vague character of the principles involved in the subject, a corresponding degree of latitude might naturally be expected in their practical application. Accordingly we find, instead of that uniformity which is so desirable, a diversity so great as almost to discourage any hope of its removal. The only mode in which this could be attempted with any probability of success would be by a sort of compromise, all parties consenting to adopt a medium for the sake of common advantage. The average pitch and depth of the various threads, used by the leading engineers, would thus become the common standard, which would not only have the advantage of conciliating general concurrence, but would, in all probability, be nearer the true standard for all practical purposes than any other.

Some years ago Messrs. Whitworth & Co. altered the threads of their screwing tackle on this principle, and the result of the experiment has proved abundantly satisfactory. An extensive collection of screw-bolts was made from the principal workshops throughout England, and the average thread carefully observed for different diameters. The ¼, ½, 1, and 1½ inch were particularly selected, and taken as fixed points of a scale by which the intermediate sizes were regulated, and the only deviation made from the exact average was such as was absolutely necessary to avoid the great inconvenience of small fractional parts in the number of threads to the inch. The following table shows the number of threads to the inch, standard measure, for each diameter.

DIAMETER IN INCHES.	THREADS TO THE INCH.	DIAMETER IN INCHES.	THREADS TO THE INCH.
3/16	24	2	4½
¼	20	2¼	4
5/16	18	2½	4
⅜	16	2¾	3½
7/16	14	3	3½
½	12	3¼	3¼
⅝	11	3½	3¼
¾	10	3¾	3
⅞	9	4	3
1	8	4¼	2⅞
1⅛	7	4½	2⅞
1¼	7	4¾	2¾
1⅜	6	5	2¾
1½	6	5¼	2⅝
1⅝	5	5½	2⅝
1¾	5	5¾	2½
1⅞	4½	6	2½

It will be observed that above one inch diameter the same pitch is used for two sizes. This was unavoidable without introducing small fractional parts; moreover, the economy of screwing apparatus is promoted by repetition of the thread.

Farther, it is important to remark that the proportion between the pitch and the diameter varies throughout the entire scale. Thus the pitch of the ¼ inch is one-fifth of the diameter; that of the ½ inch, one-sixth; of the 1 inch, one-eighth; of the 4 inch, one-twelfth; of the 6 inch, one-fifteenth. It is obvious that more power is required as the diameter increases; but this consideration alone will not account for the actual deviation, which is obviously much less than it would be if the scale were calculated mathematically with reference to the power required. The necessary amount of power must be determined in relation to the muscular force of the human arm, aided by the leverage of the screw-key. Now in the case of smaller screws, there is a considerable excess of force, and consequently of power. Again in the larger we discover a deficiency of power, for with all the leverage that can generally be applied, it requires the united force of several men to fix a bolt of six inches diameter. Hence it is evident that at the two extremes of the scale the amount of power required is not the leading consideration in determining the pitch of the thread, and in the smaller sizes the necessary depth of a coarser thread, as already observed, would too much weaken the centre part of the screw. It may also be mentioned that coarse threads would render small screws apt to work loose for want of sufficient hold to prevent the effect of jarring; and, on the other hand, finer threads on large bolts, besides being weaker, and consequently less durable, might render it a matter of difficulty to unfix them when occasion required.

It may, perhaps, be necessary to remark that the threads, of which the preceding table shows the average, are used in cast as well as wrought iron, and this circumstance has, doubtless, had the effect of rendering them somewhat coarser than they would have been if restricted to wrought iron. The variation in depth among the different specimens, before alluded to, was found to be greater proportionally than in pitch. The angle made by the sides of the thread will afford a simple and convenient expression for the depth. The mean of the variation of this angle in one inch screws was found to be about 55°, and this was also very nearly the mean of the angle in screws of different diameters. As it is obviously desirable that this angle should be constant, more especially with reference to general uniformity of system, the angle of 55° has been adopted throughout the entire scale; a constant proportion is thus established between the depth and pitch of the thread. In calculating the former, a deduction must be made for the quantity rounded off, amounting to one-third of the whole depth—that is, one-sixth from the top, and one-sixth from the bottom of the thread. Making this deduction it will be found that the angle of 55° gives for the actual depth rather more than three-fifths, and less than two-thirds of the pitch. The precaution of rounding off is adopted to prevent the injury which the thread of the screw, and that of the taps and dies might sustain from accident.

Two descriptions of tools are employed for cutting screws by hand; namely, the screw-plate and the screw-stock with moveable dies. The first, and doubtless the most ancient form, is simply a flat plate of steel, assuming the shape of a file, having a tang and handle at one or both ends; in this plate are one or more series of graduated screwed holes, so that by passing the bolt or pin successively through several a finished screw is obtained. This form of tool, however modified in its construction, is obviously imperfect, and but rarely used except for screws under $\frac{3}{8}$ inch diameter.

The first decided improvement with which we are acquainted is due to Mr. Peter Keir, who introduced a cutter, let into a groove sunk in one of the dies, which follows the lead obtained by the dies, and deepens the thread. This arrangement is more especially applicable to square-threaded screws.*

In 1828, Mr. J. Jones submitted to the Society of Arts of London† an improved form of screw-stock and tap, for which he received the thanks of the society. In this case, also, a cutter is used, secured by clamps on the face of the screw-stock, which necessarily follows the lead obtained by the dies, and completes the screw in an expeditious manner. The altered form of tap is a combination of the taper and plug tap, the part towards the point being conical, and the upper part cylindrical. The threads are rounded off both at top and bottom, and the tap is fluted with four or more rectangular grooves, one side of which is in a line with the centre, thus giving, in a cross section of the tap, a form somewhat similar to a ratchet-wheel. About one-third of the threads have their tops filed down to diminish the quantity of surface in contact, by which much labour is saved, as the greater part of the power requisite for screwing in the usual way is expended in overcoming the friction, and not in cutting away the superfluous metal. This form of tap answers perfectly well for nuts not exceeding one inch and a quarter, but for those of larger size, as two or three inches, it is advisable to insert a cutter in the body of the tap just at the part where the cone terminates, by which nearly the whole of the metal is cut out, and the upper or plug part of the tap has nothing to do but to equalise and smooth the thread.

In 1838 M. Gouet proposed a new form of screw-stock with four dies, two of which were conductors or guides, and the other two acted as a screw-cutting or chasing tool. In the *Bulletin de la Société Industrielle de Mulhouse,** we find a description of two forms of screw-stock, and an expanding tap by M. Lamorinière. The first is composed of three dies, two of which are of tempered steel, and the third of wood, intended merely to serve as a conductor. The second has four dies, very narrow and directly opposite to each other, which are made to approach by means of a circular plate, hollowed in an elliptical shape, and its circumference cut into teeth. With regard to the expanding screw-tap the object of the inventor appears to have been to dispense with a series of taps of different sizes, to cut rather than press out the metal, and to allow sharpening on a grindstone when the cutting edges become impaired.† M. Waldeck also invented a screw-stock with a series of cutters, which produced either angular or square threads, and the same mechanician subsequently introduced farther improvements with regard to taps.‡

The screw-stock invented and patented by Mr. Joseph Whitworth, of Manchester, next claims our attention. Of this tool there are two forms, the first is rather complicated: the dies, of which there are three, work in as many eccentric curves sunk in a metal disc, whose exterior edge is cut into teeth in the manner of a tangent-wheel, and worked by an endless screw, the action of which necessarily causes the dies either to approach or recede from a common centre.||

This tool cuts the metal with great rapidity, requires but little exertion, and produces very excellent screws. The principal objection to it, is that the complication of its parts, and the wear and tear of the tangent-wheel render frequent repair necessary, more especially in the hands of careless or indifferent workmen.

The second, or guide screw-stock, is entirely new in form, and not liable to the same objection; moreover, it is alleged by the inventor that it will cut a screw scarcely inferior to that obtained in a slide-lathe from a true *guide*. The thread produced is not only true, and of the exact pitch required, but perfectly formed throughout, being cut clean, without distortion of the metal.

In all these respects the advantage of the guide over the common screw-stock is remarkable. The latter will not cut a screw in any degree perfect; the thread, besides being irregular, is never of the right pitch; it is also more or less swollen by the violence done to the metal, so that the diameter of the screw is frequently found to exceed that of the blank bolt in which it is cut. These defects are attended with the most serious practical inconvenience; they frequently render it extremely difficult to obtain a fit between the screw and the nut, and consequently occasion a considerable sacrifice both of time and labour. They necessarily impair, in a very great degree, the efficiency of the screw-bolt, which cannot

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* Gill's Technical Repository, London, 1826, vol. viii. p. 193.

† Vol. xlvii. London, 1829, p. 135.

* No. 89. 1844.

† See Bulletin de la Soc. d'Enc. Tome 39, for a description of this tap.

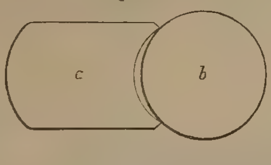
‡ See Bulletin de la Soc. d' Enc. Tome 37, 39, for farther particulars.

|| The arrangement here described is the same as that adopted in More's double Screwing Machine, and represented by fig. 6, Plate XXXII.

possess either the strength or mechanical power which it would have if the thread were cut clean and true.

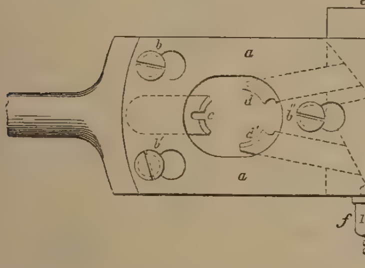
The defects in question are variously modified according to the size of the master-tap used in cutting the dies. If they have been cut by a master-tap double the depth of the thread, larger in diameter than the bolt to be screwed, they will act very well at first, and the thread will be started true, but, as the operation proceeds, they become altogether unsteady and uncertain in their action. If, on the other hand, they have been cut by a master-tap of the same size as the bolt to be screwed, the thread is made out of truth in its origin. They first touch the bolt only on the extreme point of their outer edges, as shown in the annexed sketch, *a* being the die, and *b*, the pin or bolt. Farther, they have neither sufficient guide nor steady abutment till the operation is on the point of completion. It is not unusual to employ a master-tap of an intermediate size. In this case, however, it is obvious that the dies will combine in a modified degree, the defects peculiar to each of the cases already mentioned. In the guide-stock this perplexity is entirely obviated, and the dies act with full advantage from the commencement of the operation to its conclusion. They are cut by a master-tap double the depth of the thread, larger in diameter than the screw blank; while their general form and the direction in which they are moved forward, are such as to preserve their cutting power, and steadiness of action, undiminished to the full depth of the thread.

Fig. 21.



The plan of the guide-stock will be easily understood from the subjoined sketch. The interior of the stock is shown in dotted lines through the top plate *a*, which is fastened by the screws *b*, *b*,′ *b*″; *c* is a stationary or fixed die; *d*,*d*′ are moving dies simultaneously brought up by a piece *e*, sliding in a recess in the stock, and bearing with a distinct incline, as shown by dotted lines, against the back of each die. The piece *e* terminates with a square-threaded screw *e*′, and is drawn up by a nut *f*, on the outside of the stock. The dies having been cut by a full-sized master-tap, as already mentioned, the curve made by their outer edges is that of the blank pin or bolt they are intended to screw. Hence, in starting the thread they bear at all points of the common curve, and the impression made by indentation is an exact copy of the thread of the die. The parts indented serve as a steady guide to the dies in cutting round the blank pin. A groove in the stationary die facilitates the operation. Four cutting edges are brought into action, at points of the circumference nearly equidistant; so that by little more than a quarter turn, the thread is completely started round the pin, and the difficulty involved in the operation, by the common screw-stock, is entirely removed.

Fig. 22.



After the thread is started, the fixed die serves principally as a guide and abutment for the others. The moving dies are peculiar both in regard to their form and direction, which depend on the position of the arc in the shank of the die. The two sides have each a different inclination to the arc. As the die moves forward one side becomes prominent towards the screw-bolt, and its cutting edge continues in contact with the thread, till it is cut to the full depth required. The prominent sides of the moving dies are those turned towards each other.

The direction of the common die in screw-stocks of the old form, is necessarily towards the axis of the screw-bolt. In the guide-stock the direction of the moving dies is that of two planes meeting beyond the centre of the stock, in a line parallel to the axis of the screw-bolt, and considerably behind it.

This direction is determined by reference to the change which takes place in the relative position of the screw-bolt as the thread is cut deeper. One of the three dies being stationary, there must necessarily be a constant change in the position of the screw-bolt in relation to the two others, the effect of which, if not counteracted, would be to deprive the cutting edges of the requisite prominence, but by giving them the direction before mentioned, the proper degree of prominence is secured, notwithstanding the change of position, and the latter when combined with the eccentricity of the dies, so far from being any impediment to their action, materially assists it. The newly-formed thread is thereby kept in contact with the dies, for some distance behind their cutting edges, affording them the same kind of support throughout the operation which they have at the commencement; when, as already observed, the curve made by their outer edges, is coincident with that of the screw-blank. This continued support which is necessary to steady their action, could not be obtained without a change in the position of the screw-bolt. They would otherwise acquire too much clearance as they form the thread deeper, and their cutting edges would be apt to dig.

The steadiness of the guide-stock, and its easy action in screwing, are equally remarkable. In using it, not one-half the force consumed by the common stock is required. The inner edges of the moving dies, which principally act in cutting out the metal, are filed off to an acute angle; this enables them to cut with extreme ease, and without in any degree distorting the thread, while they take off shavings similar to those cut in the lathe; their action in cutting is in effect the same as a chasing-tool, to which indeed they bear an obvious resemblance in form, and they may be sharpened on a grindstone in the same manner.

A practical difficulty has hitherto attended the use of the screw-stock, arising from the wear of the taps and dies. The tap becomes less in diameter, and consequently taps the hole too small, while the opposite effect takes place with the dies, which, being unable to cut a full-sized thread, leave the screw too large. The only mode of counteracting this two-fold error, so as to obtain a fit between the screw and the nut, is by forcing the dies forward till they have reduced the diameter of the screw a proportionate quantity, and from what has been before observed, it is manifest that this cannot be done in the case of common dies, without injury to the thread. In using the guide-stock, on the contrary, it is attended with no disadvantage, and lest the diameter of the screw should inadvertently be reduced more than necessary, figures are stamped on the sides of the nut *f*, to indicate when the thread is full.

We have now to describe Mr. Bodmer's patent screw-stock. This tool is constructed on the principle of the ordinary screw-stock, with such additions and alterations as appeared necessary. The principal objection to the old form is that the metal is rather pressed out than cut, with an expenditure of much force; in that now under consideration, one die acts as a guide, and the other as a cutter, by which arrangement not only is a perfect thread produced, but the tenacity of the metal is preserved and less power employed.

Fig. 23.

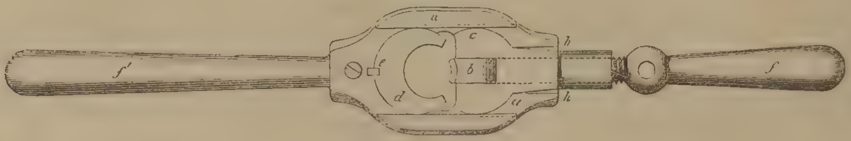
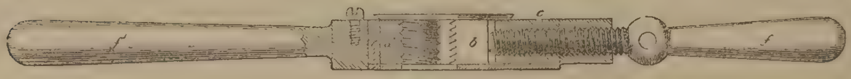


Fig. 24.



Figs. 23 and 24 show Mr. Bodmer's improved screw-stock, with the lid removed in a plan and longitudinal section; *aa'* is the box made either of steel, wrought, or cast iron; *b* the vibrating tool or cutting die, which is fixed in the die-holder *c*, in such a manner as to accommodate itself to the inclination of the thread when the die begins to cut on the surface. The die may also be a perfect fit in the die-holder *c*, but in that case it must be cut to a larger diameter than the screw itself would require, as usually done in common stocks; *d* is the guide die recessed into the stock *aa'*, and which may be bored out to the full diameter of the bolt or pin to be screwed, or tapped in the ordinary manner. The guide die *d* is prevented from turning by a small key *e*; the screw *f*, in the die-holder *c*, is not only the handle or lever by which the stock is worked, but also advances the cutting die *b*, as the operation proceeds. The cutting die-holder *c* is fitted into a circular recess in the stock *aa'*, and has as much room at *h* and *h* as is necessary to allow that part of the cutting die which, when the stock is turned in the opposite direction, would drag, to recede out of the thread so as to clear the thread and particles of metal cut out during the operation, by which arrangement the cutting die will preserve its keen edge. Suppose the operation of screwing to have been commenced at the bottom of a pin, and the stock arrived at the top; the handle or screw *f* will require to be advanced a little, and then the stock is ready to work in the opposite direction. It is evident that the moment the handles *f*,*f'*, are pulled by the workman, the die will bite on that side which is moved deeper by the pull, and recede out of cut on the opposite side; it will therefore act and cut like a tool in a lathe or planing machine, and preserve its keen edge much longer, and remove filaments of metal much more easily than dies constructed in the ordinary way.

Fig. 25 is an end view of the stock; fig. 26 a ground plan and an end view of the cutting die-holder, and fig. 27 the lid of the stock fitting the bevil or half V grooves of the same; fig. 28 is a plan and section of the guide die *d*, and fig. 29 shows a mode of regulating the play or motion of the cutting die *b*, by means of the set screws *g*,*g'*.

Fig. 25.

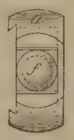


Fig. 26.

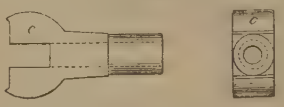


Fig. 27.

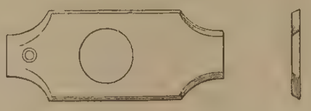


Fig. 28.

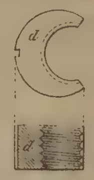


Fig. 29.

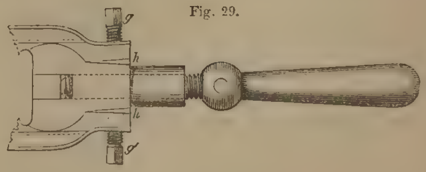


Fig. 30 is a ground plan, and fig. 31 a section of a stock with two cutting dies moving in a lateral direction; *ad* is the stock or frame; *b*,*b'* the handles or set screws, acting upon the blocks *c*,*c'* which are perfect fits in the stock, and against which the cutting dies *d*,*d'*, slide laterally. These dies are confined between two plates which are screwed or riveted to the stock in the ordinary manner. It is evident that the two cutting dies *d d'*, when tightened up against the piece which is to be screwed, will recede in the contrary direction to that of the pull, as much as there is space left between the dies and the side of the stock, and in so doing will operate in the manner already described with reference to the vibrating dies.*

Fig. 30.

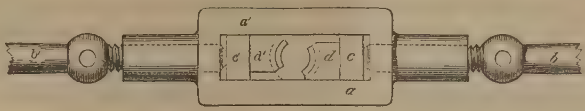


Fig. 31.

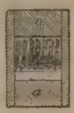
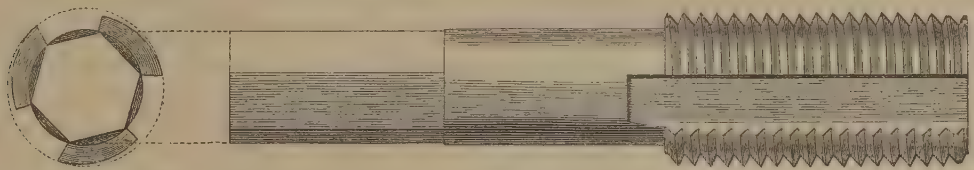


Fig. 32 shows a longitudinal and end view of one of Mr. Bodmer's convolute taps. After having been finished to nearly the right measure in the screwing lathe, the taps are subjected to the operation of a mechanism in a tap-cutting lathe, by means of which the convolute form is given.

Fig. 32.



The advantage of this construction of tap is evident, because not only is the top of the thread eased in a convolute form, as usually done by hand, but likewise the bottom: the sides of the thread also are tapered, or relieved, in the same proportion, so that the tap cuts like an ordinary turning tool, instead of making its way through the metal by sheer pressure.

The following table indicates the number of threads per inch both for angular and square threads.

V THREADS.		SQUARE TAPER THREADS.	
DIAMETER IN INCHES.	THREADS PER INCH.	DIAMETER IN INCHES.	THREADS PER INCH.
$\frac{5}{16}$	18	$\frac{5}{16}$	9
$\frac{3}{8}$	16	$\frac{3}{8}$	9
$\frac{7}{16}$	14	$\frac{7}{16}$	8
$\frac{1}{2}$	12	$\frac{1}{2}$	7
$\frac{9}{16}$	11	$\frac{9}{16}$	7
$\frac{5}{8}$	11	$\frac{5}{8}$	7
$\frac{11}{16}$	10	$\frac{11}{16}$	7
$\frac{3}{4}$	10	$\frac{3}{4}$	6
$\frac{13}{16}$	9	$\frac{13}{16}$	6
$\frac{7}{8}$	9	$\frac{7}{8}$	6
$\frac{15}{16}$	8	$\frac{15}{16}$	6
1	8	1	5
$1\frac{1}{8}$	7	$1\frac{1}{8}$	4
$1\frac{1}{4}$	7	$1\frac{1}{4}$	4
$1\frac{3}{8}$	6	$1\frac{3}{8}$	3
$1\frac{1}{2}$	6	$1\frac{1}{2}$	3
$1\frac{5}{8}$	5	$1\frac{5}{8}$	$2\frac{3}{4}$
$1\frac{3}{4}$	5	$1\frac{3}{4}$	$2\frac{1}{2}$

To describe, within the limits of a brief essay, the various tools used among the different classes of turners is manifestly impossible. They are so infinitely diversified both in form and size according to the necessities, the ingenuity, and frequently, perhaps, the prejudice of those who use them, that a volume would scarcely suffice to do justice to the subject.

We shall therefore content ourselves with simply indicating a few of the main features of that class of tools, without occupying our space with lengthened descriptions of their individual peculiarities; and, in the first place, we may remark that *gravers*, *triangular*, *square*, *round*, *pointed*, *heel* or *hook*, and *screw tools*, with various other nameless sorts, the contrivance of individual skill, are used in turning hard bodies, such as bone, ivory, and the metals.

The graver is made from a square bar of steel cut off by an oblique plane at the end, which forms a lozenge or diamond face, and produces two inclined edges, at two of the flat sides of the bar; these are inclined opposite ways, so that the graver serves either for left or right hand work by merely turning it one quarter round to bring up another side. The point formed by the acute angle in which the two inclined edges meet, is better adapted for cutting than any other form, and is exceedingly strong; the flat sides give it an excellent bearing upon the rest. Another convenience of the graver is the ease with which it may be sharpened, an object of considerable importance in turning hard metal; it only requires to be held on the grindstone at the proper angle to grind the diamond face away, and thus make sharp edges with the two flat sides. The graver is principally used to rough the work, its point being applied to cut grooves all over the surface till it is true, and then the plane edge of the graver, or a square, or round tool, makes it smooth, and of a proper figure.

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* Although only two methods of applying oscillating or vibrating dies are given, Mr. Bodmer does not confine himself to these identical arrangements, because the manner of causing the die to move in and out of cut during the operation of screwing, may be contrived in various ways. For instance, the guide die *d*, shown in fig. 48, may be made to move instead of being stationary, and the cutting die may be fixed, or the die itself may vibrate on a pivot, or there may be one or two sliding dies, as shown in figs. 30 and 31, moving in a lateral direction. The principle claimed by the inventor is that of placing the cutting die during the operation of screwing, and by the action of the stock itself, as it is turned in the one direction or the other, alternately into such a position as to bring one of its cutting sides, or series of cutting edges, into cut, while the other recedes as much out of cut, as to drag on the metal.

Triangular and square tools are so denominated from their respective sections being of these figures; they are flat at the end. The former have three cutting edges, namely, each arris in a longitudinal direction; the latter, which are principally used for turning brass, have four, that is, each arris at the extremity.

Round tools have the edges of a semicircular form, and are used for forming hollow mouldings, and for roughing wood.

The pointed tool has two inclined edges, forming a point, which cuts grooves in any piece of work, or its edges may be used to turn shoulders either right or left.

Heel-tools are used for turning wrought iron, steel, and copper; they are made with all the edges already described, but the end where the edge is formed is bent, so that when it is presented to the work in a proper direction, the handle is inclined upwards in such a position, that the end of it will rest upon the workman's shoulder, and he holds it down firmly with both hands, the heel of the tool being at the same time supported on the lathe-rest. The metals above mentioned, being of a fibrous texture, turn away in a connected shaving; the tools are therefore presented in the direction of a tangent to the work, but as the drift of the work would force the tool endways, it is necessary to have a heel or angle which is placed immediately upon the rest, then the long handle serves to guide and fix it, and by elevating the end its edge cuts deeper.

Cast iron is turned by hook-tools; their edges are formed in various ways, but invariably very obtuse, being nearly a right angle; in turning they are held in such a position, that a line bisecting the angle of the edge is made to point nearly to the centre, and as the metal is usually hard and refractory, they are made with a hook which, being laid over the rest, acts as a lever, and causes the edge to approach to, or recede from the work by merely raising or depressing the end of the handle.

Screw-tools are very important, and in most cases indispensable appendages to a lathe; they are usually made in pairs: namely, an outside and inside tool, and the teeth of both should be so accurately cut, that on being placed together, the teeth of the one in the intervals between the teeth of the other, they should exactly fit, even to the exclusion of light.*

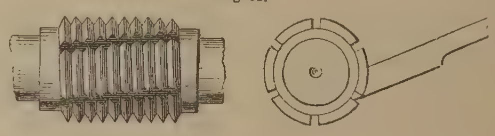
Many methods have been suggested to enable the mechanic to cut his screw-tools. M. Séguier, a distinguished amateur turner, recommends that a model be taken in lead or soft brass, by impression of the required screw, and then to place the pattern so obtained and the blank tool back to back in a vice, and with a triangular file remove the steel, until the projecting teeth exactly coincide with the model. To this we object that the form of a triangular file does not agree with the shape of a screw-thread; it is much too obtuse; what is called a slitting file is certainly more suitable. With very great care and dexterity in the use of the file, this method may answer, but the operation demands an aptitude and precision of hand rarely attained—added to which the loss of time and risk of failure are scarcely compensated by the probability of success.

We now proceed to explain a method very generally adopted to cut the teeth in screw-tools. A piece of cast steel is turned cylindrical, and being suspended between the centres of the lathe is made to revolve; upon the surface of this cylinder a series of concentric and equidistant circles are cut by means of a screw-tool, or a simple V-tool, which is held firmly, and in a fixed position, against the steel cylinder. When the teeth are sufficiently raised by the cutting action of the tool, the cylinder is removed from the lathe, and gaps or notches cut across its surface in a diagonal direction, so as to give to the teeth a cutting edge. It is then hardened, and tempered to a straw colour. This is technically called a *hob* or *hub*.

The great objection to this method is, that it produces perpendicular and not inclined teeth in the screw-tool. This, however, is easily remedied by cutting a regular helix, instead of merely concentric circles upon the surface of the hob; or still more readily by employing a common plug-tap, which answers the purpose perfectly well.

We will now suppose either a hob or plug-tap to be made to revolve between the centres of the lathe. The workman takes a blank screw-tool, which must be well annealed, and applies its face to the revolving hob; being careful to hold the tool very firmly, yet not to allow the hob at the commencement to bite too greedily, and supplying oil to the surface of the hob or tap, which essentially assists the operation. The blank tool may be held either above or below the centre of the hob. The latter is shown in fig. 33, and is in some respects preferable to the former, as it affords a better purchase for the tool. The method practised in Manchester of cutting screw-tools is in many respects similar to that we have just described, except that it requires the aid of change-wheels and a slide-lathe. Nevertheless, as many of the details are common to both, the observations we are about to make will, in some measure, apply to manual as well as mechanical power.

Fig. 33.



The first thing is to cut the hob or hub, which is effected by a self-acting slide-rest. It is simply a screw cut on the surface of a solid cylinder of cast steel, with diagonal grooves cut across the thread of the screw to act as cutters, as shown in fig. 33; the two necks of the hob have concave holes drilled in the ends to carry the centres of the lathe.

The hob is placed between the centre points of the lathe, by means of a dog or catch attached to one end in the usual way. Change-wheels are then put on to connect the mandrel or spindle with the guide-screw of the lathe, and which carries along the slide-rest. The wheels are so arranged that one turn of the mandrel* causes the slide-rest to travel

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* It may probably appear fastidious to insist on this rigid perfection in a tool which is apparently of simple and easy construction; a little consideration however will show that unless the teeth, whatever be their shape, are similar in every respect, it will be impossible to cut an accurate thread, since if one tooth be in the least degree larger than the others, it will necessarily destroy the proportion of the thread.

* It is scarcely necessary to observe that for each different pitch of a screw, the change-wheels and hob require to be altered.

a distance exactly equal to one thread. The blank which is to be cut, is firmly screwed down in the tool-box of the slide-rest, and made to stand above the centre of the hob, as shown in fig. 34. It is then pressed, by the screw of the slide-rest, against the hob; and the lathe being put in motion, causes the tool to traverse along and against the hob, cutting it as deep as may be thought necessary. The face of the tool, when cut, is a segment of a circle varying, of course, according to the diameter of the hob. Fig. 35 is a side view of the tool in this condition; but this form is not found sufficiently economical in practice, since it can only be ground and sharpened to a particular point, as to *b*, for when ground to *c*, as from *a* to *c*, it ceases to cut, owing to the top of the tool being then as far from the screw to be cut as the bottom. The method adopted to obviate this difficulty is to give the tool an angular instead of a circular face, and this is managed in the following way: the screw-tool is removed from the slide-rest, and as the hob revolves the workman elevates and depresses the end of the tool which is in his hand, so as to present different points of the face to the cutting action of the hob, until by degrees he succeeds in obtaining a perfectly angular face, which allows the tool to be ground nearly or quite to the bottom, with a certainty of preserving a good cutting edge. In order to fix the blank which is intended to make an inside screw-tool, in the slide-rest, some little contrivance is necessary, the stem is usually bent, and afterwards, when cut, set straight previously to hardening.

Fig. 34.

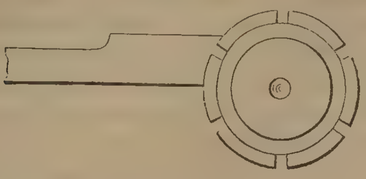
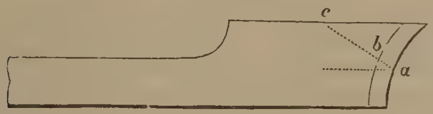


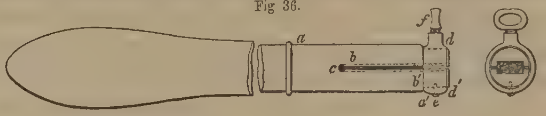
Fig. 35.



The handles of turning tools, it may be observed, must be varied in size, according to the manner in which they are intended to be held. For heavy work, more especially when the lathe is turned by machinery, they must be sufficiently long to reach to the shoulder, upon which one end rests during the operation of turning, besides being held by both hands of the workman at the same time. In using the foot-lathe the tools are held by both hands only, and the handles are rarely more than half the length required in the former instance.

Mr. Holtzapffel's socket handle for turning tools, fig. 36, is an extremely ingenious and useful appendage to the lathe as it is equally applicable to slide-rest tools. This handle is 9¾ inches long, the brass socket *a a′*, has a longitudinal slot *bb′*, which terminates at the circular hole *c*. This socket is confined by the steel ring *d d′*, which has at one side a steel set screw *e*, and at the other a pinching screw *f*, which necessarily contracts the aperture, and consequently grips the tang of a tool placed within it. The slot *bb′*, as well as the opening for the tang of the tool, and the pinching screw, are more clearly shown in the end view.

Fig 36.



Mr. Nasmyth's tool-gauge is a very simple and convenient method of ascertaining whether a tool is ground or formed to the proper angle. It consists of a planed plate of metal, on whose surface there is fixed at one end a conical steel pin, whose taper, or the angle formed by the sides of the cone with the surface of the plate, is exactly that which is proper for the cutting face of the tool. The angle formed by the sides of the cone and the surface of the base plate, should be about three degrees. By using this gauge, all difficulty of forming the tools to the proper angle is at once removed. Moreover, this same gauge will answer for every kind of planing or turning tool of whatever size.[*]

We now proceed to a very important practical inquiry, namely, the velocity at which tools cut most advantageously for different kinds of material.

It cannot, we apprehend, have escaped the observation of those of our readers who are in the habit of turning metal, that if a velocity exceeding certain prescribed limits be imparted to the material, the edge of a cutting tool applied to reduce the surface of that material is brought to a soft state and rendered obtuse. This is an acknowledged fact, and many ingenious contrivances have been, from time to time, introduced to meet the exigency of the case, or in other words, to regulate the speed of the lathe-mandrel according to the hardness and diameter of the metal or other substance to be turned. It is commonly supposed that for wood the velocity cannot be too great, yet this is probably a vulgar error, since if we allow the speed to pass certain limits, the tool necessarily becomes hot, loses its temper, and ceases to cut. Wrought iron requires a slow motion, and cast iron, above all, ceases to be affected by the edge of the tool, unless a very slow and regular motion is preserved, as it appears to act by abrasion, and actually grinds away the face of the tool.

The opinion of practical men is much divided on this point —some name from ten to fifteen feet per minute as a *maximum* velocity, others allow thirty to forty feet, whilst others again regard this as the *minimum* speed which should be given to cast iron, in order to obtain the greatest effect from the tool.

For turning or boring cylinders or, indeed, any substance of which the diameter is nearly equal throughout, a uniform velocity fully answers the purpose, since the speed can easily be increased or diminished by means of conical pulleys placed in opposite directions, as also by many other mechanical contrivances* which are too familiar to practical men to require any extended description on this occasion. Suppose, for example, we have a cylinder of wood of considerable diameter to turn, we devise some method to control the speed, as for instance, by diminishing the diameter of the driving pulley, and increasing that of the pulley on the mandrel; by this means the motion is made slower, and in some respects proportioned to the diameter of the material. Had the cylinder been composed of cast-iron instead of wood, we should have pursued a somewhat similar course, but carried to further limits than that we have just described. The facility afforded by two elongated cones, fixed in opposite directions, as regards their respective diameters—the one forming the immediate connection with the driving power, and the other on the lathe mandrell—enables us to regulate the velocity with a precision that in many operations is of the highest importance. But this arrangement is too limited to afford by itself all the variety of speeds required for general work. The back-geer lathe, Plate XLI., answers the purpose perfectly well for cylindrical turning. In this arrangement the driving cone *b* and pinion *p* are connected, and may either run loose or be locked to the spur-wheel *w*, but in the former case the speed of the lathe spindle, and consequently whatever is attached to it, is greatly reduced, as indeed is manifest from inspection of the engraving; for supposing the spur-wheel *w*, and pinion *p*, to have respectively 52 and 13 teeth; and the spur-wheel *h*, and pinion *i*, on the back-geer spindle, also 52 and 13 teeth, the ratio of speed of the driving cone to that of the lathe spindle, would be as 16 to 1; or, in other words, the latter would perform 1 revolution for every 16 revolutions of the former.

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[*] See further, Manuel du Tourneur, par L. E. Bergeron, seconde édition par P. Hamelin—Bergeron, Paris, 1816, 3 tom. 4to.

L'Art du Tourneur par M. Paulin Desormeaux, Paris, 1824, 2 tom. 12mo, avec atlas des planches.

Nouveau Manuel du Tourneur, rédigé par M. Dessables, nouvelle édition, augmentée. Paris, 1837, 2 tom. 24mo.

Rees's New Cyclopædia, 4to. Art. *Turning*.

Gill's Technical Repository, vol. ix. *passim*. Lond. 1826, 8vo.

M. Armengaud,† a distinguished French engineer, has arranged the different degrees of velocity applicable to the several mechanical operations of turning, boring, drilling, &c., in the following manner.

The velocity at the circumference of the material, or of the tool for turning or planing cast iron, should be from 2$\frac{3}{4}$ to 3$\frac{1}{4}$ English inches per second.

The velocity of the tool at its circumference for widening or broaching cast-iron is about 1$\frac{3}{4}$ inches per second.

The velocity of cast iron, turned by a hook-tool, held and guided by the hand of the workman, to finish or complete is 4$\frac{3}{4}$ inches per second.

For iron turned by means of the slide-rest, the velocity at the circumference of the work is about 5$\frac{1}{2}$ to 6 inches. When the metal is turned by hand tools, the speed at its circumference is from 7 to 8 inches to rough out, and from 11 to 12 inches to finish.

The difference of velocity of the work or the tool when the turning is effected mechanically or by the hand, is deduced from the obvious fact, that in the former case the contact between the tool and the work is constant and invariable, whilst in the latter it is intermittent.

The foregoing velocities are equally applicable to drills or cutters in boring machines.

The lateral progress of the tool varies according to the power of the machine; it may be stated in general at about ·0125 inch for each revolution of the work, but it should be less for drills.

The following table indicates the average degree of speed, as well for turning as boring.

DIAMETER IN INCHES.	REVOLUTIONS OF SPINDLE PER MINUTE.	DIAMETER IN INCHES.	REVOLUTIONS OF BORING-BAR PER MINUTE.
1	50	1	25
2	25	2	12.5
3	16.67	3	8.33
4	12.50	4	6.25
5	10	5	5
6	8.32	6	4.16
7	7.15	7	3.57
8	6.25	8	3.125
9	5.55	9	2.77
10	5.	10	2.5
15	3.33	15	1.66
20	2.50	20	1.25
25	2	25	1
30	1.667	30	0.833
35	1.430	35	0.714
40	1.250	40	0.425
45	1.12	45	0.56
50	1	50	0.5
60	0.834	60	0.417
70	0.716	70	0.358
80	0.626	80	0.313
90	0.554	90	0.278
100	0.50	100	0.25

If we have occasion to turn a plane surface accurately true, the motion of the lathe-mandrel, or what is the same thing, the substance affixed to it, requires to be accelerated or retarded in a ratio proportioned to the progress of the tool, either to or from its centre; then that portion of the plane where the tool takes effect, would pass its edge always at the same velocity; and if a proper speed be obtained in the first instance, not only will the tool preserve its originally keen edge for a very considerable time uninjured, but the surface produced by its action would be nearly perfect. This control and command of the movement obviously require that it be continuous; since if the lathe be stopped, a mark or false cut, as it is termed, will be the unavoidable result.

Under ordinary circumstances, regular motion in surface-turning is not only prejudicial in relation to its effects, but it also involves great waste of time. We will suppose that a speed suitable for the circumference and the proximate parts is obtained, it is evident, as we approach the centre, the rotatory movement becomes less effective; until at length near the centre it produces little or none, and the work does not proceed at all. The reason of this is obvious; the velocity continues unaltered, while the diameter of the material is progressively reduced. It is also manifest as an inevitable consequence of uniform motion in surface turning, that presuming a suitable velocity to be communicated when the tool is at the greatest distance from the centre of rotation, if it be made to advance regularly towards the same point, similar uniform speed being continued, the cutting edge of the tool would not, on its arrival at the centre, be more deteriorated than if the velocity had been increased in a proportionate ratio to its progress towards the centre. This fact is sufficiently manifest; but, it must be observed, that by maintaining a uniformity of the angular velocity, we sacrifice nearly one-half the time employed in doing the work. This statement admits of an easy geometrical illustration. Thus: let the parallelogram, *abcd*, fig. 37, represent the time that would be required to turn a surface. Draw the diagonal line, *b c;* bisect the line *a c*, at *e; e c*, at *g;* and *g c*, at *i;* then draw the lines, *e f*, *g h*, and *i j*, parallel to *a b*.

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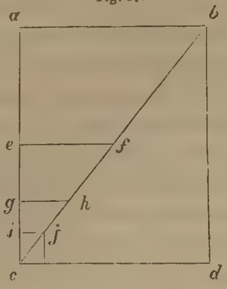
* Those who wish to investigate the subject will gather much information from the specification of Mr. Bramah's patent, long since expired. See also the article in Rees's Cyclopædia on *Expanding Riggers*.

† In the Glasgow Practical Mechanic and Engineer's Magazine, vol. iii. pages 287 and 379, will also be found a most valuable practical paper on the subject of cones and lathe speed-pulleys, pointing out a method for facilitating their calculation under all circumstances. L'Ouvrier-Mécanicien, Guide de Mécanique Pratique, Par Ch. Armengaud, jeune, 2nd edition, Paris, 1843, 12mo.

Fig. 37.



Let *c* represent the centre, *a c* equal the radius, and *a b* equal the circumference, or time of one revolution at its greatest diameter; therefore, the lines *e f*, *g h*, and *i j*, will also represent their circumference, or time of one revolution at their respective radii at *e*, *g*, and *i;* and as the lines *a b*, *e f*, *g h*, and *i j*, are one-half the length of each other, so will their revolutions be performed in similar proportions of time, and the velocity of the lathe-mandrel will be increased in the inverse ratio, as the length of the lines *a b*, *e f*, *g h*, and *i j;* consequently, the right angled triangle *a b c* will represent the time that would be required to turn a surface, when the velocity of the lathe-mandrel is increased in the manner already described. The parallelogram *a b c d* will represent the time that would be required, if the velocity remain unaltered—that is, from the moment the tool is applied to the surface at its greatest diameter, to its arrival at the centre; for if the length of the line *a b* represent the time of one revolution at its greatest diameter, the line *c d* will similarly represent the time of one revolution when the tool reaches the centre; therefore, as the length of the line *c d* is equal to *a b*, so will all the intermediate revolutions be performed in similar spaces of time.

This inquiry may be usefully applied to determine the period of time necessary for surface-turning. Thus if we wish to know what time would be required to turn a plane surface of cast iron, the diameter being twenty-four inches, to make fifty revolutions or cuts in each inch of the radius, and to pass the tool at the rate of 15 feet per minute? Multiply the circumference = 75·39 inches by the radius = 12 inches, then multiply the product = 904·68, by the number of revolutions or cuts in one inch of the radius, in this instance 50, this will give 45234 inches; divide this by 12, to reduce it to feet, and we have 3769·5; divide again by 15, which gives 251·3 minutes, and lastly dividing by 60, we have 4 hours, 11·3 minutes; consequently this would be the time, if each revolution were performed in equal portions of time, but if the speed of the lathe-mandrel be regulated so that the surface to be turned shall always pass the tool at the same velocity, then the time required to perform the work will be only one-half of the above; for in this case, we must multiply the radius by one-half of the circumference, as that will be a mean proportion of the lengths of all the intermediate revolutions.

We have now arrived at a very important and interesting inquiry:—namely, the principle and mode of action of automatic or self-acting tool-machines.

If we consider the separate and distinct parts which combined, make up a machine, whether simple or complex, to be disunited and viewed in detail, we shall find that their constituent parts, however numerous, are composed either entirely or partially of certain original geometrical figures, so that it is evident the more nearly the configuration of each individual part approaches strict mathematical truth, the more regular and perfect will be the performance of the machine.

But the accuracy and precision of workmanship here predicated, and which peculiarly distinguishes the machinery of the present day, is obviously unattainable by mere manual dexterity; it is principally to be attributed to the *slide-rest.*

The invention and introduction of this tool may justly be considered an era in the history of constructive mechanism; it has entirely superseded both manual labour and dexterity, which previously were required; added to which, it enables us to produce work infinitely superior, and in a much shorter space of time, than can be effected by hand-turning. So many and so conclusive are the beneficial results consequent to the introduction of this tool that it is not affirming too much to assert that nearly all the improvements in modern machinery are in a greater or less degree to be attributed to its almost universal application in some or other of its many and varied forms.

It constitutes no part of the present inquiry to investigate the principles of turning, except so far as is absolutely necessary to illustrate the subject in hand. Let us suppose then, that instead of the tool being held and guided by the hand of the workman, assisted merely by muscular strength, the same cutter is firmly secured to the lathe-rest, so that during the operation of cutting, it can be slid along the bed of the lathe, in a direction parallel to the axis of the work, the result of this operation will necessarily be a cylinder; if, however, the tool move in a line forming an angle with the axis of the mandrel, a conical form will be obtained, and if it operate at right angles to the same axis a plain surface will be the result. Such are the elementary principles on which this important, and in many respects invaluable tool is constructed, the details of which, and the different forms, we defer for the present, thinking it preferable in the first instance to notice the machines of which it forms an appendage.

One of the primary and most indispensable requisites of a well constructed lathe, is that the centre of the cone-spindle should coincide exactly with the adjustable centre of the moveable head stock, or, in other words, that each of these parts should be in the same line, parallel to the face of the lathe bed.

The spindle is a very important part of the lathe, as upon its truth and accuracy of motion the circular rotation of any work attached to it mainly depends; it is usually made of iron, but the working parts of the two extremities are altogether of steel, which are hardened after being turned and finished; they are then ground in their places to fit the collars or bearings with finely pulverized Turkey stone and oil; the left-hand end has a hole bored exactly in its centre to receive the point of a screw, which supports and retains it in its place, as shown in Plate XLI. figs. 1, 2, and 3.

The other, or right-hand end of the spindle, is somewhat larger, and has a conical hole bored in the direction of its axis for the purpose of receiving a centre point; this is disengaged when necessary by means of any tapered instrument which, being inserted in a slot cut in the mandrel, acts as a lever and forces the centre forwards.

In the larger class of lathes the spindle is usually fitted up to run in divided collars of brass or gun metal; these slide on V-shaped grooves cast in the head stock, and are adjusted to fit the neck of the mandrel or spindle by means of screw bolts which pass through a cap or plate of wrought iron, fitted on the upper surface or top of the head stock.

Various contrivances have from time to time been suggested to avoid the inconvenience of a back screw, and at the same time insure uniformity of position under all possible circumstances. The boring and turning machine, Plates XLVII. XLVIII., offers an example of this modification; here the spindle works in divided collars but has shoulders at the necks, by which contrivance all longitudinal motion is entirely avoided. The method adopted by Mr. Whitworth of Manchester to effect this object is extremely ingenious, and peculiarly entitled to the distinctive epithet, self-sustaining; as a specimen of the adaptation of mechanical means applied to produce certain results, it is probably unrivalled.

In this lathe the inner journal of the spindle is turned conically, but at two different angles, that part next the nose being more acute than the remaining portion. This arrangement meets the great difficulty attendant on conical bearings, as the base of the cone is opposed to direct pressure, and consequently removes all danger of the spindle becoming fixed or jammed in its collar. The sliding cone which is placed upon the spindle for working in the outer bearing, becomes, as it were, a part of the spindle, but having longitudinal motion; it tends to balance the effect of pressure applied directly to the screw.

The method shown in the sectional view, fig. 10, Plates XLIX. L., is by no means so well contrived as in this case; a set screw is introduced to counteract longitudinal motion by direct pressure against the left-hand end of the spindle.

The moveable or right-hand head stock now requires consideration. The first decided step towards improvement in this part of the lathe, may doubtless be traced to the substitution of a plain cylinder for a screw, both terminating alike in a conical point; in the former case, the cylinder turning upon itself, the apex of the cone is not so liable to deviation, but this does not compensate for the absence of regular pressure, which can be imparted at will, and at any moment, by means of a screw, and that in a degree exactly required by the nature of the case; nevertheless this great inconvenience must not be overlooked, the screw is extremely liable to vary relatively to the centre line. If, for instance, either the pin or the thread be slightly twisted, which it frequently and unavoidably is, the point will describe a small circle at each revolution, and however well and carefully the thread of the screw may be cut, it is almost impossible to make it perfectly regular and mathematically true; consequently, if the central line of the cone, which forms the point, be not that of the screw itself, the same effect will be produced.

These and other considerations of a like nature that might be adduced, probably suggested the arrangement shown in Plate XLI. fig. 5, which combines the simplicity and uniformity of position of the cylinder with the mechanical power of the screw. In this case the sliding cylinder in the head stock is deprived of rotatory motion, its outer end being connected by a coupling to a second cylinder of smaller diameter, which constrains both to move in a parallel direction, —between these, a square-threaded screw works, which is capable only of longitudinal motion, being connected to the aforesaid coupling, so that the screw being worked to the left hand, compels the cylinder in the head stock to travel with it, as is evident from inspection of the engraving.

The form of head stock just described probably suggested that generally known as the cylinder head stock. This was invented simultaneously, we believe, by M. Collas of Paris, and Mr. Joseph Clement of London.* The arrangement now usually adopted, is shown in section fig. 38. The head stock is bored out true and an iron or steel cylinder ground therein so as to insure an accurate fit; this receives a forward and backward motion from a screw which is rendered endless by means of a collar or cap, and is worked by a handle wheel. The sliding cylinder is fixed when requisite by a pinching screw, which presses against a piece of iron let into the head stock. A sectional view of another mode of fitting up is given in fig. 39.

Fig. 38.

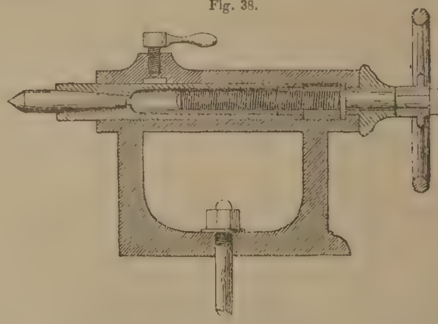
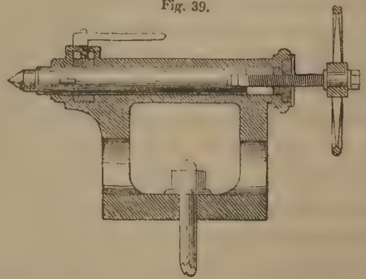


Fig. 39.



This is unquestionably not so expensive to get up as that just described, but it is liable to the objection, that nearly the whole amount of pressure is thrown upon the driving screw connected with the cylinder and attached to the handle wheel. In this latter example the mode of fixing the spindle in its required position is superior to that shown in fig. 38. A malleable iron ring, bored out so as accurately to fit the spindle, is let into a recess in the head-stock, and tightened up by means of a handle passing through a screwed shank projecting from the ring.

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* For a full account of the first see Journal des Atéliers, tom. I. pp. 167-171. The second is described in L'Industriel, No. 41.

In some cases it is desirable to possess the means of moving the shifting head-stock in a direction at right angles to the bed of the lathe. A very convenient mode of effecting this is shown in Plates XLIX. L., by means of the screw *f*, which causes the head-stock to move transversely, an arrangement which is peculiarly applicable to conical turning.

In heavy lathes the sliding head-stock is usually moved along the bed by means of a train of bevil wheels, as in Plates XLVII. XLVIII., fig. 4. Here a bevel pinion attached to a horizontal spindle, which is worked, when necessary, by a crank handle fitted upon the square end *o*, gives motion to a bevil wheel upon one end of a vertical shaft, which has its bearings inside a hollow column cast in the body of the head-stock for that purpose. On the other end of this shaft is a bevil pinion; this again geers with a small bevil wheel keyed upon the spindle *p*, which works in bearings attached to the sole of the head-stock and also carries a pinion which works into the rack M, fixed upon the bed plate of the lathe, and thus obviously completes the connection, enabling the workman to adjust the sliding head-stock in any required position with ease and facility.

Cones or speed-pullies are very important adjuncts to tool-machines in general, and more especially the lathe, as from the nature of the operations performed by it, it is a primary requisite that the range of variation in the velocity of the spindle should be as large as possible. Professor Willis[*] has investigated the mechanical principles of their adjustment in a very clear and satisfactory manner.

Supposing a pair of cones or speed-pullies to be arranged upon two parallel axes and in opposite directions, we have an easy mode of changing the ratio of the angular velocity of the shafts by simply moving the belt from one pair of speeds to another.

In this case, it is evident that the diameters of each pair of opposite pullies should be so adjusted that the belt shall be equally tense upon any pair of the whole series; this, as may be easily demonstrated, is attained by making the sum of each pair of opposite pullies equal throughout the whole series.[†]

We have now to describe the slide-rest, a tool which has unquestionably contributed more than any other to the improvement of modern machinery. The invention of this truly important tool is claimed by Mr. Nasmyth for the late Mr. Henry Maudslay; but, as it appears to us, the conclusion arrived at by that gentleman has been hastily adopted, and without sufficient inquiry, inasmuch as a form, certainly similar in all important details, was well known, and commonly used by rose-engine turners, long previously; added to which we remark that the original slide-rest[*] constructed by Mr. Maudslay for Mr. Bramah bears so slight a resemblance to that now in use as scarcely to be capable of identification; moreover, it is extremely doubtful whether a form of rest, known as the parallel rest, as well as a tool very similar in principle, invented by the late Earl Stanhope for turning metallic surfaces of large dimensions truly plane, did not precede the slide-rest of Mr. Maudslay.

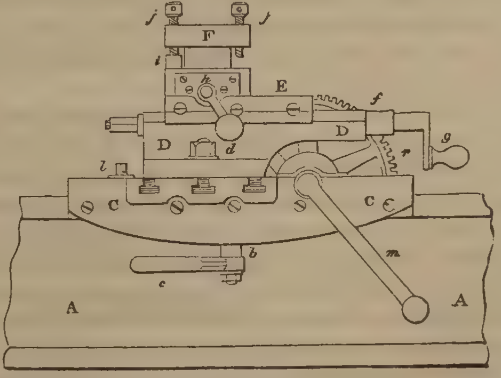
It is foreign to our purpose to pursue the subject further, nor are we disposed to depreciate the great merit of an ingenious and highly talented engineer, nevertheless as the principle has been so extensively and so successfully applied to modern tool-machinery, we have been solicitous to perform an act of justice in attributing to those who have in any way contributed to the invention of this important tool, a fair share of commendation.

The form of slide-rest shown in Plates XLVII. XLVIII. is a very convenient arrangement in many of its details, and the one most commonly adopted. Here J is the saddle-plate upon which the slide-rest K is supported, and the longitudinal slide L which carries the tool-holder is firmly secured upon the part K by the screw *u;* the parallel motion of the tool-holder is obtained by means of the screw *r*, and similarly a transverse motion of the same in order to place the tool in and out of cut by the screw *s*, these necessarily work at right angles to each other, and the tool itself is made fast on the tool-holder by the two clamps *u*. The adjustment requisite for setting the tool to the work is effected by disengaging the sole K from the saddle-plate J, the nuts on the bolts which pass through the slots, shown in the plan view fig. 2, being released, by this means the sole K may then be moved to the requisite distance from the longitudinal axis of the lathe, and to a certain extent in the line of that axis by shifting the bolts in the dovetail grooves of the saddle-plate, should that operation be more convenient than to shift the latter on the bed-frame of the lathe.

A transverse adjustment to a limited extent may be obtained by means of the screw *s*, and the tool-carrier may be adjusted longitudinally by the screw *r*.

The slide-rest we have now to describe is due to the ingenuity of Mr. Joseph Whitworth. It is unquestionably an

Fig. 40.



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[*] Principles of Mechanism, by R. Willis, M.A., F.R.S. Lon. 1841. 8vo.

[†] Strictly speaking, this proposition holds good only in cases where the belt is crossed.

[*] See Buchanan on Mill Work, edited by George Rennie, Esq., Ap. B. p. 401. Mr. Rennie himself refers to Plumier's Art de Tourner en perfection, 1750, fol. See Preface, p. xl.

excellent specimen of constructive mechanism, combining the requisite stability with great accuracy of motion, and the manner in which the details are worked out displays considerable ability and mechanical talent. Fig. 40 shows the rest set for parallel turning; as, for example, an iron rod in the direction of the axis of the lathe. Figs. 41 and 42, the latter being a section, show it set for facing a circular plate; that is, the motion of the upper slide is in a line at right angles to the lathe-bed AA, which is bolted in the usual manner to the supports BB.

The saddle upon which is placed the carriage of the rest is simply a broad and strong plate of cast iron CC, planed true and finished by scraping upon both horizontal faces. The slides *aa*, similarly planed and dressed, are screwed to the lower surface of the saddle on either side of the lathe-bed to enable it to traverse with uniform motion its entire length without shake or play, and so arranged as to compensate for wear and tear by means of lateral screws countersunk in the cheeks of the saddle CC.

When the saddle is required to remain stationary during the work, as, for example, when a circular or other plate is to be faced, it is firmly fixed on the lathe-bed AA by means of a single screw-bolt *b*, the nut of which is screwed up by the lever handle *c*.

The carriage of the rest is composed of three principal parts; the first DD, which rests upon the saddle, and is susceptible of different positions; the second E is a plate moveable upon the preceding, and the third F which carries the tool. The base or carriage is of cast iron, planed and finished by scraping not only on the two horizontal faces, but also on the two upper lateral edges, which are angular like those of the lathe-bed, so as to receive the slides *ee* fitted on each side of the rectangular plate E. A screw-bolt *d*, the square head of which is lodged in a T-groove in the saddle CC, serves to adjust the carriage DD upon the saddle-plate, and this adjustment obviously depends upon the diameter of the piece to be turned.

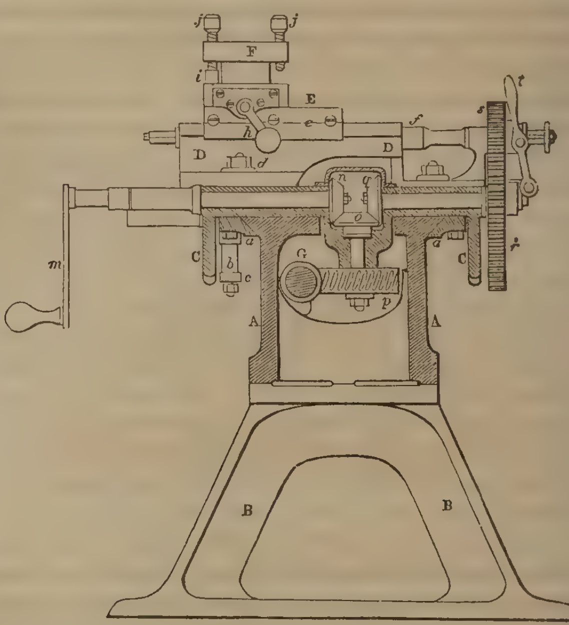
The rectangular plate E is rendered moveable in the direction of the length of the carriage DD by means of an endless screw *f*, which is entirely sunk in its thickness, and receives a rotatory motion either from the handle *g* or from the lathe itself. This endless screw works in a brass nut fitted under the plate E, and being deprived of endlong motion, it necessarily follows that in its rotation it imparts a forward and backward motion to the nut, and consequently to the tool-carrier.

The part F, which may properly be designated the tool-carrier, is susceptible of another movement—that is, in a direction exactly at right angles to that just described—by means of a second endless screw smaller than the former; this is intended to be worked only by the handle *h*, and that in the event of its being necessary to regulate in an exact manner the position of the tool with regard to the piece upon which it is to operate. This endless screw works in a brass nut attached to the plate E, and consequently imparts a forward and backward motion to the tool-carrier F, which moves between two slides screwed upon this plate.

The tool *i*, which is intended to act upon the material either for turning or screw-cutting, is securely fixed on the tool-carrier F by vertical pinching screws *j*, *j*, which are screwed through the thickness of the upper plate or cap; these screws, four in number, are placed at the angles of the cap, an arrangement which allows the tool to be fixed in different directions, and in such a manner that it is always acted upon by two screws. This disposition enables the workman to employ two tools which shall act at the same time or nearly so upon the material; for instance, the one to rough out and the other to complete the work.

Motion is communicated to the carriage, and consequently to the tool-slide, by a peculiar arrangement of the guide screw, which is so formed as to be alike capable of performing the office of a rack as well as that of a screw; to this end the thread is rounded off both at top and bottom, instead of being either triangular or square. It is thus enabled to work either in a nut or with a tangent wheel.

Fig. 41.



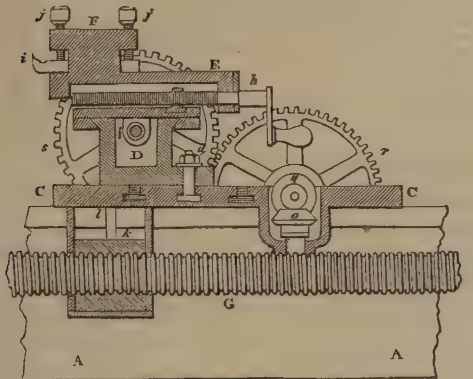
This guide screw is shown at G, figs. 41 and 42, placed within the lathe-frame, not in the direction of the axis of the machine, but rather on one side, in order to screen it from the falling turnings; the nut *k* is taken out of geer, when necessary, by the pin *l*. When the guide screw is required to answer the purpose of a rack—as, for instance, to bring the saddle, and consequently the carriage, to any particular place—it is done by disengaging the nut *k*, and turning the handle *m*, which is fitted on a horizontal spindle carrying the small meter wheel *n*, which geers with a similar one *o*, keyed on one end of a vertical shaft placed in the centre of the saddle C. On the other extremity of this shaft is a tangent-wheel *p*, which works with the guide screw of the lathe. Now as this last is deprived of rotatory motion, it is evident that by turning the handle *m*, the tangent wheel *p*, driven as already described, will produce precisely the effect of a rack; that is to say, the saddle and carriage will receive a traversing motion in the direction of the length of the lathe-bed.

In actual work the saddle, and consequently the rest itself, being placed in the required position, the handle *m* is removed and the nut *k* brought into connection with the guide screw G, which, actuated and regulated by a train of wheels attached to the lathe, causes the saddle with its appurtenances to travel with any degree of speed that may be required.

By a peculiar and ingenious arrangement, the guide-screw is made to drive the carriage and tool-carrier in a direction at right angles to the axis of the lathe. This is effected by means of a combination of small bevil and spur wheels *n*, *o*, *q*, *r*, *s*. The wheel *o* geers with the two marked *n* and *q*, having their axes in the same straight line. On the opposite extremity of the axis of the wheel *q* is fixed the spur wheel *r*, which geers with a similar spur wheel *s* mounted on an iron spindle, which terminates at the other end in a grooved shoulder; this axis is movable in a socket, which forms a support, and is fixed to the saddle C, and by means of an ingenious contrivance the forked lever *t* is made to connect or disconnect at pleasure the spindle that carries the spur wheel *s*, with the square end of the screw *f* of the carriage DD.

It is obvious, if we suppose the saddle to be fixed on the lathe-bed—and to effect this it is merely necessary to screw up the bolt *c*—that the guide screw G giving motion to the tangent-wheel *p* determines the motion of the toothed wheels, and consequently that of the screw *f*, which after this manner gives motion to the carriage and the tool-carrier, to which we have given, by anticipation, the position shown in the sectional view, fig. 42. It is evident that when this transverse motion is not required, it is only necessary to throw the wheels out of geer by means of the forked lever *t*, and then these wheels will revolve on their axes without producing any effect.

Fig. 42.



The collars or bearings in which the axes of the bevel-wheels *n* and *q* revolve freely, are nothing more than long hollow cylinders bored out true, and fixed on the saddle or bed-plate *c*, and to avoid the injury which might result from these wheels becoming clogged by chips of metal, they are usually protected by a metallic cover either of tin or sheet-brass.

In the construction of steam-engines and engineering work generally, there are a great number of parts, such as steps, bushes, &c., which require their outer diameter to be turned truly concentric with the hole bored through them. The most general method of accomplishing this, is by driving the work upon a mandrel sufficiently tight to withstand the action of the turning tool. The common mandrel, which is perhaps the most universal adjunct of the lathe, is a cylindrical bar of iron turned with an exceedingly slight taper to fit the central hole of the work.

The time lost in preparing these mandrels, and the great weight of useless metal which must thus be kept in stock, prove serious objections to their use, and led Mr. Hick to the invention of the expanding mandrel, by which various sizes of holes may be fitted.

Figs. 43 and 44 represent a longitudinal section and an elevation of the mandrel, the expanding wedges being shown in two different positions. *a* is the mandrel, the central portion of which is turned conically as at *b*. This cone is provided with four dove-tail grooves *c* running in the direction of the axis of the mandrel, and fitted to receive the four wedges *dd*, shown in fig. 43 in their highest position. The dotted circles in the end view represent the work, which is placed upon the four wedges; these are pressed onwards by the hollow conical collet *e* urged by the nut *f* working on the screw threads cut on the mandrel. In this manner the wedges *d* are driven up the inclined grooves, and thus fix the mandrel concentrically within the hole of the work so that any diameter of hole may be readily fitted, which is within the range of the travel of the wedges.

Fig. 43.

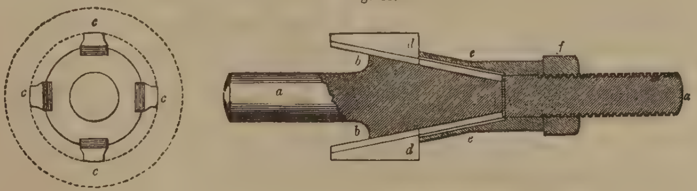
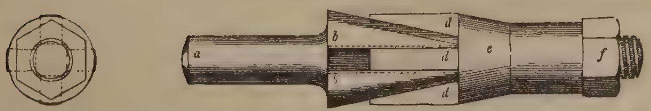


Fig. 44.



Another equally important appendage of the lathe, is the universal chuck. Various views of this chuck, as made by Mr. Charles Walton, are given in Plates XLIX., L. For turning or boring articles of a regular external configuration, this arrangement has a decided advantage over the common chuck, where each adjusting screw is moved separately; and effects a considerable saving in time, in setting the work.

There are, besides the modification just referred to, various other species of chucks, amongst which we may class Mr. Bodmer's as one of the best.

In this arrangement, the clutches are expanded and contracted by means of a series of radiating screws, each of which carries a pinion geering with a large central wheel on the front plate of the chuck; the work is fastened by setting the lathe in motion, and holding back the front plate until the wheel upon it shall have driven in the clutches worked by the screws, sufficiently far to grasp it.

The object of change-wheels applied to a lathe, is, generally speaking, to obtain a screw of any required pitch; that is, in relation to the leading or guide screw* by which the cutter is moved in a longitudinal direction.

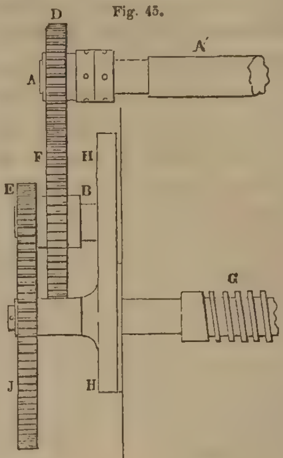
If a spur-wheel be attached to the left-hand end of the lathe-spindle, and so arranged as to geer with another spur-wheel similarly fixed on the axis of the guide-screw, and continuous motion be communicated to the lathe-spindle, it is evident that this motion will be transferred by means of the aforesaid wheels to the saddle of the slide-rest; consequently a screw-tool attached thereto, will receive direct rectilinear motion, and thus trace the spiral thread of a screw on the exterior surface of any revolving cylinder opposed to its action.

The relative proportions of these wheels obviously determine the pitch of the screw to be cut, as compared with that of the guide screw of the lathe; so that if they are of equal diameter, or, what is the same thing, have an equal number of teeth, the result will be a screw of the same pitch as that of the leading screw; but if the driving wheels be larger than the driven—suppose in the proportion of two to one, then will the pitch of the work be exactly double that of the guide screw, and if these proportions were reversed a contrary result would follow.

It is obvious in such a case as we have here supposed, that, as the wheel fixed on the lathe spindle and that upon the guide screw, each revolve in contrary directions, all screws cut by this arrangement will be the reverse of the guide screw, or left hand threaded. In order, therefore, to cut a right hand screw, it will be necessary to introduce an intermediate wheel geering with both wheels, so that the direction of motion of the work shall be the same as that of the lathe spindle.

The principle of this arrangement is shown in fig. 45, entirely disconnected from the frame-work of the lathe, and without strict regard to proportion, it being intended merely to exhibit the parts as distinctly as possible. Here A A′ is a portion of the lathe-spindle, to which is attached in the usual way a cylindrical rod, for the purpose of cutting a thread upon it. G represents the leading or guide screw revolving in suitable bearings, and giving motion by means of a nut to the saddle, and consequently the carriage of the rest, upon which is firmly clamped a suitable tool intended to cut the screw.

Fig. 45.



In this arrangement it is manifest that every revolution of the guide screw G will cause the rest to advance through a space exactly equal to its own pitch, or, in other words, supposing the guide screw to have four threads in the inch, it will in every revolution it makes, advance the rest, and consequently the tool or cutter, one-fourth of an inch endlong upon the work, so that if the lathe-spindle revolve with the same velocity as the guide screw, the tool will produce a screw of precisely similar pitch; but if on the contrary A A′ revolve with less velocity than G, then the effect will be a greater pitch, and *vice versa*. Now if the lathe-spindle and the guide screw be connected by a set of change wheels, we have the means, by properly choosing the numbers of these wheels, to obtain any desired pitch. This is practically effected by an intermediate axis which is supported by a grooved bearer; this carries an arrangement of additional change wheels, according to circumstances and the conditions of the case. The mode of action is as follows:—the leading or guide screw which communicates motion to the saddle of the slide-rest is driven by a train of wheels which are in connection with the spindle of the lathe; it passes through and forms the axis of a moveable piece H, and at its extremity carries the fast wheel J which geers with a pinion E; this and the wheel F, which geers with the pinion D upon the end of the lathe spindle, are carried by a stud B fixed in a straight slot cut in the moveable arm H, which has likewise a curvilinear slot near its end, through which two fixed studs pass; upon these studs pinching nuts are placed, which being screwed up tightly, retain it securely, and by altering the angular position of H, a pinion of greater or less diameter than D may be used, and consequently the motion of the leading or guide screw regulated.

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* Screw-cutting and boring machines are reducible, observes Professor Willis, to the principle of aggregate motion. For the cutting of a screw is in fact the tracing of a spiral upon the surface of a cylinder, and the motion of boring is also the tracing of a spiral upon the surface of a hollow cylinder; the tool being in both cases the describing point, and the plain cylinder the surface. Now as the tracing of this spiral is resolvable into two simultaneous motions, one of revolution with respect to the axis of the cylinder, and the other of transition parallel to that axis, we have in the construction of machines for boring and screw-cutting the choice of four arrangements:—

(1.) The cylinder may be fixed and the tool revolve and travel. This is the case in all simple instruments for boring and tapping screws, in machines for boring the cylinders of steam-engines, and in engineers' boring machines.

(2.) The tool may be fixed and the cylinder revolve and travel. Screws are cut upon this principle in small lathes with a traversing mandrel.

(3.) The tool may revolve and the cylinder travel. The boring of the cylinders of pumps is often effected upon this principle.

(4.) The cylinder may revolve and the tool travel. Guns are thus bored, and engineers' screws cut in the lathe.

Principles of Mechanism.

Having now explained the arrangement of geering necessary for effecting a change of speed in the guide screw, we shall, for the sake of a practical illustration, give determinate values to the wheels D, F, E, and J. Thus let the number of teeth in the wheels D and E be 30, and that in F and J 60 each, the pitch of the guide screw G being $\frac{1}{2}$ inch, or in other words, that it has two threads per inch. It is now evident that one complete revolution of G will advance the tool through the space $\frac{\text{inch}}{2}$, and similarly one revolution of A will advance the tool through the space $\frac{30 \times 30}{60 \times 60} = 0.25$ turns of G, or $\frac{1}{8}$th inch, and consequently the pitch of the screw cut by this arrangement will be $\frac{1}{8}$th inch. In this manner any desired pitch of screw may be cut by proportioning the change wheels accordingly. This may be much facilitated, by arranging the various pitches of screws in a tabular form and placing the respective change wheels required for each, opposite to them, so that all computation during the actual progress of the work is avoided.

In order, however, to meet emergencies, it is necessary that the process of calculation for any given pitch should be thoroughly understood, and for this purpose we shall give an example of the method.

Suppose it is required to cut a screw which shall contain 13 threads in the inch. Here the ratio of speed between the cone spindle and the guide screw is required to be as $\left\{\begin{matrix}13\\2\end{matrix}\right\}$ to 1; so that we have $\begin{matrix}(\text{J})\\(\text{D})\end{matrix}\frac{130}{20}, \frac{156}{24}$, &c. $= 6\frac{1}{2}$. In this case, the wheels D and J are supposed to be geered together merely by a single carrier wheel, but as this arrangement is not always convenient, we shall now find the ratios of the wheels as given in fig. 45, where four are used. Here we must remember that the condition of the case is that the numerator divided by the denominator of the expression (13) shall be $6\frac{1}{2}$. We will assume 28 and 56 as the respective values of D and J, or $\frac{\text{J}}{\text{D}} = 2$. Hence we have only to find such values of F and E, as that $\frac{\text{E}}{\text{F}} = \frac{6\frac{1}{2}}{2}$ or $3\frac{1}{4}$, which informs us that E must have $3\frac{1}{4}$ times as many teeth as F. Suppose then F has 32 teeth, we have $32 \times 3\frac{1}{4} = 104 =$ the number of teeth in E, the whole set of wheels standing as follows, D = 28, J = 56, F = 32, E = 104. This result is capable of verification as follows: $\frac{56 \times 104}{28 \times 32} \times 2 = 2 \times 3\frac{1}{4} \times 2 = 13$, or the number of threads per inch of the screw to be cut. Thus, in all cases of calculations of this nature, the expression in general terms stands thus:

$$\frac{\text{No. of teeth in J}}{\text{No. of teeth in D}} \times \frac{\text{No. of teeth in E}}{\text{No. of teeth in F}} = \frac{\text{No. of threads per inch of screw to be cut.}}{\text{No. of threads of guide screw.}}$$

The following table calculated by Mr. Whitworth of Manchester, shows the train of wheels to be used in cutting screws varying in pitch from 1 to 70 threads in the inch; the leading or guide screw is supposed to have two threads per inch, yet may the table be still employed where the leading screw has four threads to the inch, for the same train of wheels would suit for cutting screws of double fineness; and similarly when the leading screw has only one thread to the inch, a screw of only one-half the fineness will be produced with any train given in the table.

In the first columns it will be observed that the wheel and pinion, carried by the stud B are omitted; these not being required in cutting screws of the pitches there stated, are displaced, and a simple carrier wheel substituted for them. To facilitate this arrangement, the wheel J, on the leading screw, has the boss of its socket longer on one side than the other; so that when reversed, as in this instance, it is brought into train with the carrier wheel, placed upon the stud; and this again is placed in train with the pinion D.

No. of threads per inch of screw	No. of teeth on		No. of threads per inch of screw.	No. of teeth on				No. of threads per inch of screw.	No. of teeth on				No. of threads per inch of screw.	No. of teeth on			
	Mandrel pinion D.	Leading screw wheel J.		Mandrel pinion D.	Stud wheel F.	Stud pinion K.	Leading screw wheel J.		Mandrel pinion D.	Stud wheel F.	Stud pinion E.	Leading screw wheel J.		Mandrel pinion D.	Stud wheel F.	Stud pinion E.	Leading screw wheel J.
1	80	40	8¼	40	55	20	60	18	40	60	20	120	32	30	80	20	120
1¼	80	50	8½	90	85	20	90	18¾	80	100	20	150	33	40	110	20	120
1½	80	60	8¾	60	70	20	75	19	50	95	20	100	34	30	85	20	120
1¾	80	70	9½	90	90	20	95	19½	80	120	20	130	35	60	140	20	150
2	90	90	9¾	40	60	20	65	20	60	100	20	120	36	30	90	20	120
2¼	80	90	10	60	75	20	80	20¼	40	90	20	90	38	30	95	20	120
2½	80	100	10½	50	70	20	75	21	80	120	20	140	39	40	120	20	130
2¾	80	110	11	60	55	20	120	22	60	110	20	120	40	30	100	20	120
3	80	120	12	90	90	20	120	22½	80	120	20	150	42	50	140	20	150
3¼	80	130	12¾	60	85	20	90	22¾	80	130	20	140	44	30	110	20	120
3½	80	140	13	90	90	20	130	23¾	40	95	20	100	45	30	90	20	150
3¾	80	150	13½	60	90	20	90	24	65	120	20	130	45½	40	130	20	140
4	40	80	13¾	80	100	20	110	25	60	100	20	150	50	30	100	20	150
4¼	40	85	14	90	90	20	140	25½	30	85	20	90	52	35	130	20	140
4½	40	90	14¼	60	90	20	95	26	70	130	20	140	52½	40	140	20	150
4¾	40	95	15	90	90	20	150	27	40	90	20	120	55	30	110	20	150
5	40	100	16	60	80	20	120	27½	40	100	20	110	56	30	120	20	140
5½	40	110	16¼	80	100	20	130	28	75	140	20	150	60	30	120	20	150
6	40	120	16½	80	110	20	120	28½	30	90	20	95	65	30	130	20	150
6½	40	130	17	45	85	20	90	30	70	140	20	150	70	30	140	20	150
7	40	140	17½	80	100	20	140										
7½	40	150															
8	30	120															

Such are the general principles of screw cutting for single threads, but when it is required to cut a multi-threaded screw, it is evident that some additional apparatus will be requisite to effect the requisite exactitude of division, so as to bring in each parallel thread in its proper place.

Two separate contrivances have been devised for this purpose by M. M. Pihet, and Mr. Bodmer of Manchester. M. M. Pihet's apparatus is shown in figs. 46 and 47, the former of which is a front elevation and the latter a section in a line with the axis of the lathe. *a* is a cast-iron disc cut with a female screw to fit the nose of the lathe spindle, and on the face of this disc is fitted the division plate *b*, the central portion of which also answers for chucking the work. The circumference of this disc is divided by notches, into 60

Fig 46.

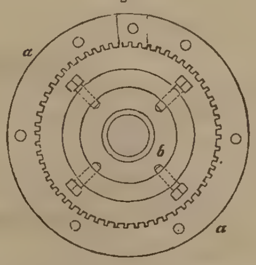
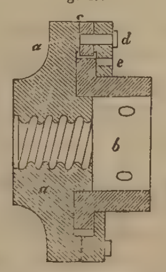


Fig. 47.



equal parts, numbered respectively from 0 to 60, into which the spring catch or stop *c* takes, so that when the disc *a* is put in motion, and the catch put down in any notch as required, the whole moves together as in one piece.

The disc *b* is held in its place merely by friction generated by the pressure of the ring *e;* this ring is shown removed in fig. 46, in order to show the graduation of the disc *e*, which is bolted to the disc *a*, and carries in one portion of its circumference a screw *d*. This screw passes through the external ring *e*, and is screwed into the stop *c*, so as to fix the latter in any required division of the plate *b*.

In cutting a multi-threaded screw by this apparatus, it takes the place of the common chuck, the workman fixing his work to it, by means of the radiating screws in the tubular portion of *b*.

Supposing it is required to cut a triple-threaded screw, the first operation which is necessary, is the detaching of the stop *c* from the disc, so that the latter may be turned round alone, until the division 0 comes opposite the hole in the external ring seen at *e* in fig. 47; the screw *d* is then adjusted so as to connect the division plate with the disc *a*, and the first thread of the intended screw is cut throughout its whole length. Now as the thread is to be a triple one, it is obvious that the circumference of the work must be divided into three equal parts, so that each thread may be equidistant when cut; accordingly the stop is again detached, and the division plate carrying the work, turned round, until the notch 20 arrives at the hole *e*; the stop is then replaced, and the second thread is cut in a similar manner as before. Lastly, the division plate is moved round, until the notch 40 is seen through the hole *e*, when the remaining thread is cut, the three having the same reference to each other as the divisions 0, 20 and 40 relatively bear to each other.

In the production of the minor screws and bolts used in engineering work, the lathe is superseded by the screwing and tapping machine, in which the thread is formed by a travelling die working upon the revolving bolt intended to be screwed. The principle of the action of this machine will be understood by referring to Plate XXX., in which we have given an elevation, ground plan, and different detailed views of a single screwing machine of great simplicity. In this arrangement the frame containing the dies, travels upon the parallel guide rods B, B, the work being fixed in the chuck F, and entered into the dies, which are then contracted until they embrace the bolt sufficiently to be drawn along its surface by its revolution. This machine is remarkable for the very ingenious method by which the dies are made to change their relative positions, while preserving their common centre in the axis of the screwing spindle; for a detailed account of which see the Description of this machine, page 24.

Fig. 48.

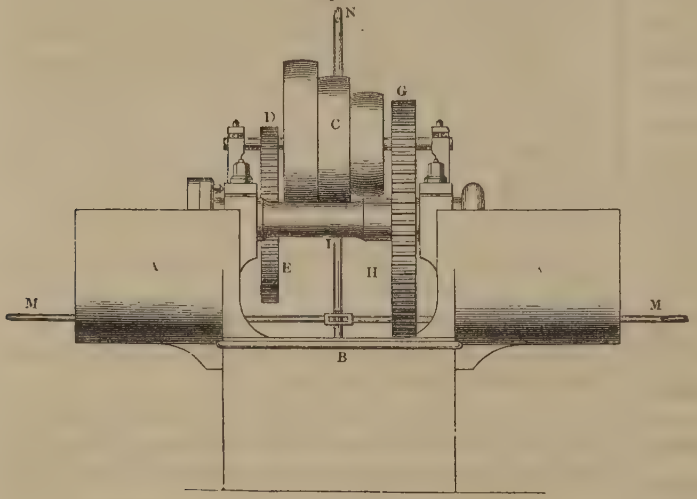


Fig. 50.

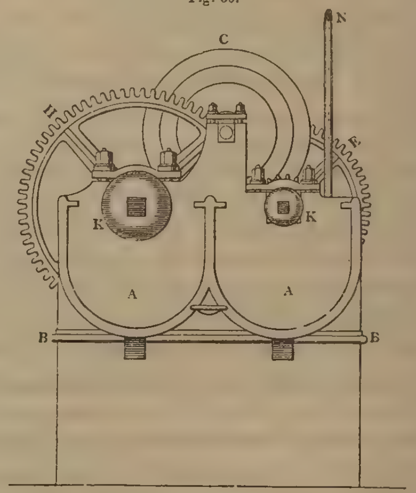


Fig. 49.

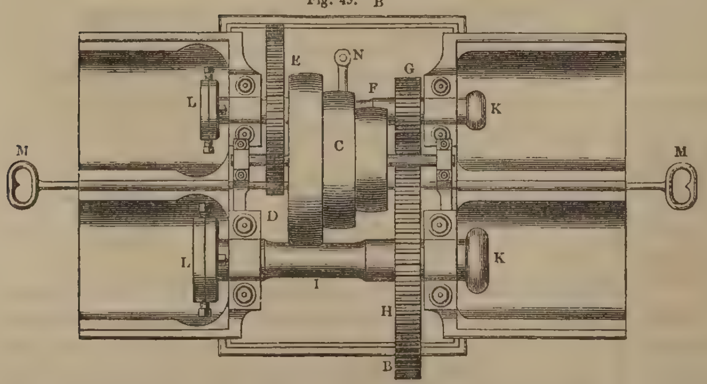
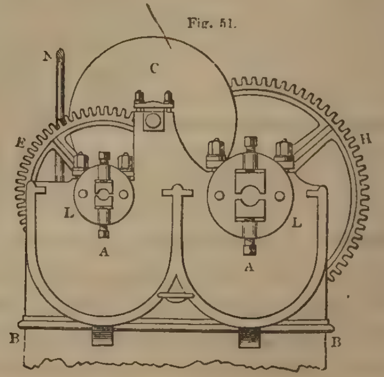


Fig. 51.



The double screwing machine by Messrs. Randolph, Elliot and Co., of Glasgow, is a very complete and useful workshop auxiliary, and, moreover, possesses the important advantage of great compactness. Fig. 48 is a side elevation of the machine, fig. 49 is a ground plan, fig. 50 is an elevation of the nut tapping end, and fig. 51 is a similar view of the bolt screwing end. The frame work of the machine AA, and the sole plate BB, are cast in one piece. The driving cone C is supported by the upright frames near the centre of the machine, and carries a pinion D, of 15 teeth, which geers with the wheel E, of 56 teeth, keyed on the smaller screwing spindle F; thus the relative speeds of the driving cone and the spindle F are as 3.73 to 1. The spindle F again carries a pinion G of 18 teeth, geering with the wheel H of 72 teeth, keyed on the larger screwing spindle I, the ratio of speed in this case being as 4 to 1. In the end view fig. 50, the chucks K,K are shown with square recesses, for the purpose of receiving the heads of taps for tapping nuts. In the view of the contrary or bolt screwing end, the chucks L,L are provided with plates grooved for the purpose of receiving the screwing dies, which are adjusted by means of the set screws, which press against the backs of the dies so as to afford the necessary adjustment. The motion of the spindles is stopped or reversed by the handles M, M, at each end of the machine. They are connected by a lever with the vertical rod E, which carries the shifting strap forks. The top driving geer consists of three pullies, a fixed central one, with a loose one on each side, all being on one shaft, which also carries a cone of three speeds exactly similar to C; the last carries the driving belt communicating directly with the machine. The centre pulley of the set of three is much narrower than that on each side of it, so as to allow of the release of the cross strap before the open one comes upon it, and *vice versâ*. For another somewhat similar arrangement see Plates XXXI., XXXII.

Allied to the subject just discussed, is that of nut-cutting, an operation formerly entirely performed by manual labour —a tedious and uncertain process, but now effected with facility and precision by self-acting machinery.

In Plate XLII. we have given detailed views of a complete machine for this purpose, by Mr. Archibald Mylne of Glasgow. The nut to be cut is fixed on the upright spindle of the support *r*, and is brought in contact with the revolving cutter *x* by means of the screw *y*, the requisite division of the faces of the nut being effected by means of the circular table, which is provided with six equal notches fitted with a spring catch from the lever *n*. This machine is also provided with a self-acting feed motion, by which the nut is gradually moved up to the cutter while in action. This is effected by the shaft *c* of the speed pulley *e*, which carries a worm *s* geering with the wheel *h*, upon the shaft of which is fixed a pinion geering with a rack on the lower side of the table carrying the nut. This motion may be dispensed with when thought necessary, and the work may be carried forward by hand, by means of the hand-wheel and shaft *b*.

This machine is very compact and fully answers the purpose of dressing nuts of any number of sides, by using differently divided plates. In many engineering works,—we may instance those of the Messrs. Stephenson of Newcastle, nuts of all numbers of sides are forged in swages made for the purpose; the accuracy and beauty of finish of which is nearly equal to nuts cut by the machine, and answer all purposes where extreme finish is not required, the expense of production being at the same time very greatly diminished.

The subject of nut cutting leads us now to the consideration of screw keys, by means of which all nuts of screws used in the connection of the different portions of machinery, are adjusted to suit the ever-varying exigences of mechanical contrivances. The common screw key with fixed jaws must be so familiar to our readers as to render any description of it unnecessary, and we shall therefore point out a few examples of attempts to remedy the defects of this most necessary instrument. In the dissection of any piece of machinery, even of the more simple species, we invariably find a multitude of different sized bolts, the nuts of which of course each require a key suited to its own particular size. A reference to the number of gradations of size of nuts, given in a previous portion of our pages, will immediately point out the necessity of some contrivance for dispensing with the number of these instruments, and introducing an adjustable apparatus, by which a number of different sized nuts may be worked without entailing the awkward drawback of keeping so many all but useless workshop appendages.

The earliest contrivance for this purpose is what is technically called a *monkey*, the use of which, previous to the introduction of the more refined species of tools, was almost universal.

This instrument is represented by fig. 52. Here the fixed jaw *a* is riveted to the extremity of the square bar *b*, the contrary end of which is screwed to fit an interior screw in the handle. This handle is provided with a double shoulder, the space between which fits loosely into the boss *c* of the sliding bar *d* which carries the moveable jaw *e* sliding on the square bar *b*. Thus the jaw *e* may be set at any required distance from *a* by merely turning the lever handle. The objection to this key is that there is no means of making up for the wear of the sliding jaw upon the bar *b*, and as a certain amount of slackness will always result after the tool has been in operation some time, it becomes exceedingly troublesome in the working. As an improvement upon this tool, the key represented in fig. 53 was invented by Mr. Stubbs of Warrington. Here the square portion of the lever handle *a* is grooved or slotted throughout for the purpose of receiving a projection from the sliding jaw *d*, which thus passes through the handle as shown at *c*; *b* is a slight screw bolted to the projection *e* of the lever, and pinned at its other extremity to

Fig. 52.

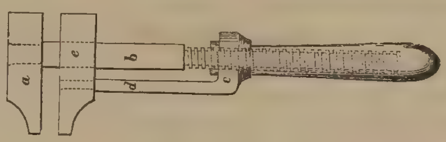
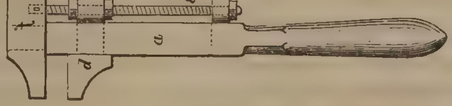


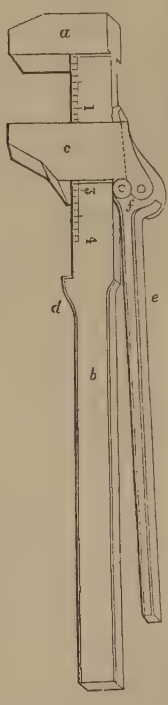
Fig. 53.



the jaw *f*. The moveable jaw *d*, when set to any particular size of nut, is held in its place by the small nuts placed one on each side of the projection *c*. In this modification, as in our last example, there is no provision for any wear of the sliding jaw, and consequently, it is so far liable to get out of order after a little use.

An extremely elegant instrument for this purpose, is the roach wrench, invented by Mr. Fenn of London, and which is equally applicable to the working of nuts and all similar purposes.

Fig. 54.



Our example of this key, fig. 54, is capable of receiving nuts from the smallest size up to four inches, and is of course sufficient of itself for all ordinary purposes. The end jaw *a* is in one piece with the lever handle *b*. The moveable jaw *c* is mortised to slide upon that portion of the lever between the end *a* and the fixed stop *d*; it is held in its place when set for any nut by means of the second lever *e*, which works on a centre pin in the projecting portion of the jaw. This lever also carries a projection at *f*, by which it is jointed to a thin wedge, passing between the top of the lever *b* and the interior surface of the slotted portion of the moveable jaw. Thus, when the latter is set to the size of nut required, a slight pressure upon the side of the lever *e* forces down the wedge, and secures the jaw immoveably.

The peculiar merit of this species of key is, that all allowance for wear is made up by the wedge, which will never permit of any looseness in the jaw, as the only difference caused by the wear of the surfaces in contact will be a greater travel of the fixing wedge. Practical men who have made use of those keys which are adjusted by means of nuts will at once see the value of this advantage.

Next to the turning lathe in its importance to the engineer, the planing machine stands foremost in the rank of constructive machines. The primary idea of planing by machinery was doubtless brought into existence by the necessity which constantly presents itself of diminishing the enormous amount of labour expended in producing plane surfaces on wood by hand, as practised by means of the common joiner's plane. Next to the process of sawing, there is no operation connected with the working of wood, which consumes so much time, and adds so much to the expense of the conversion of timber as the production of the hand-planed surface.

The first attempt to obviate this difficulty with which we are acquainted, was made by General Bentham in 1791, who took out a patent for a method of effecting this object. In this scheme, the plane or cutting edge, which was moveable, was made of the full width of the board intended to be cut, and on each side of it were fixed fillets which projected below the face of the plane, a distance equal to the amount of the thickness intended to be taken off the board. Several plans were adopted for obtaining a good surface from a very thin board, but the whole scheme eventually proved all but abortive—the machine was never practically worked by mechanical power, but whether thus driven, or by the hand of the attendant workman, the idea had still the advantage, that it exonerated the latter from the charge which he had of his tool in the ordinary operation of planing, rendering a common workman as useful as the skilful joiner for this purpose. The next epoch in the history of mechanical planing is the improvement produced by Mr. Bramah, who in 1802, patented a method of producing "straight, smooth, parallel, and curvilinear surfaces on wood and other materials." This invention embraced the original machine for producing spheres, the principle of which is still preserved in all machines of a similar nature to the present day. Bramah's planing machine, as constructed for the Royal Arsenal at Woolwich, gives us a specimen of an embodiment of his ideas at this period.

Here the cutters are attached to a horizontal disc keyed on a strong vertical spindle. This disc is put in rotation at a speed of about 90 revolutions per minute, the material to be cut being attached to a sliding cast iron bed, which is moved by hydrostatic pressure. A pipe communicating with a hydrostatic press is carried in below the bed of the machine, and terminates in a plunger barrel, the plunger of which carries a rack geering with a pinion on a rag wheel shaft. This wheel is provided with teeth, over which a pitch chain attached to the table of the machine is carried.

In all planing machines as at present constructed, the cutter is invariably the fixed portion, the work being passed beneath it in the act of cutting, by means of a sliding table. The particular species of planing machine which has been most lately introduced, is termed the hand planing machine.

In Plates XXXVII. and XXXVIII. we have given three views of a simple and effective hand planing machine, as suited for small work generally, such as links and connecting rod ends for locomotive engines, and other portions of machinery where a plane surface of small extent is required.

The table of the machine is here supported in the usual manner, as employed in similar tools of a larger class, upon a bed bolted to the top of two standards attached to the floor. The lower surface of the table carries a rack M, which is driven by a pinion F, upon the shaft G, supported in bearings attached to the fixed bed of the machine. Motion is given to this shaft in either direction, by the cross handle H, worked by hand in a similar manner as applied to small presses.

The cross slide C is supported by two uprights bolted down to the bed; this slide carries the tool holder D, which is traversed across the bed of the machine by means of the horizontal screw *b*. The automatic action of the transverse feed motion of the cross slide is effected by the moveable stud S, attached to the travelling table; this stud, being moveable in a groove in the side of the table, is capable of being set at any point in order to suit the required length of stroke for the work. The pressure of this stud, upon a short lever keyed on the small shaft carrying the piece *n*, depresses it, and the latter, by its connecting rod *r*, acts upon the ratchet plate K, upon the horizontal screw *b*.

The amount of travel thus given to the screw is varied by shifting the position of the sliding studs in the pieces *m* and *n*. The front plate of the tool holder is provided with two short circular slots, through which bolts pass from the back plate; in this manner the tool may be set at any angle to suit the nature of the work required. The tool holder or cross slide is raised or lowered, to suit the circumstances, by the vertical screws *f f*, driven by bevil geering in the usual manner.

A somewhat similar, but more useful machine of this species, has been introduced by Mr. Charles Walton of Leeds. In this machine the bed is so arranged that it may be fixed upon the workman's bench, and may be driven either by manual or steam power. Immediately beneath the bed of the machine, is placed a horizontal grooved disc driven by bevil geering either from the pulley shaft of the workshop or by a winch. This disc is grooved directly across its upper surface, for the purpose of receiving a pin, which is connected by a link with the lower surface of the short travelling table. In this manner a reciprocating motion is given to the table in the simplest manner, and the length of stroke is capable of variation according to the distance of the pin in the horizontal grooved disc, from the centre of motion. The feed motion of the cross slide is effected by a stud fixed on the under surface of the horizontal disc; this stud works a short lever keyed upon a shaft working in bearings attached to the side of the bed. This shaft again carries a second lever outside the bed, and is jointed by a link to an arrangement of ratchets similar to that in Plate XXXVII. For all small machines, this method of giving motion to the table is decidedly the simplest and most compact, and although the introduction of the disc has the effect of producing a variable speed in the cutting, being greatest at the middle of its stroke, and least at each end, yet as the disc is confined to machines of a short stroke, its diameter is not so great as to bring about a detrimental variation in the speed. Small machines of this species, which are quite an innovation in the workshop, are now becoming indispensable, where much small work is required, and have served in a great measure to banish that most expensive of all tools—the file, and thus rendered an important service in cheapening engineering work in general. So much indeed is this the case, that it is an established fact that different portions of machinery, the configuration of which is made up of curved and plane surfaces, are now entirely finished by means of the lathe and planing machine without the necessity of touching them with the file.

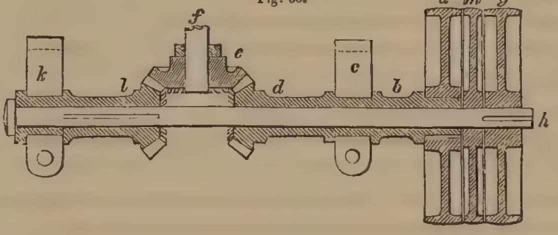
As a specimen of a step higher in the order of completeness and general usefulness in machines of this kind, we must now refer the reader to Plates XXXIX. and XL., where we have given very complete views of the machine invented by Mr. Mylne of Glasgow. Here the system of working the vertical and horizontal slides is similar to that made use of in the hand machine, Plates XXXVII. and XXXVIII.

The chief peculiarities in the present machine are, the arrangement of the geering for travelling the table, and the great completeness of the tool holder. The forward or cutting motion of the table is obtained from the large pulley A, the shaft of which carries a pinion D geering with the large wheel E, upon the rack pinion shaft.

This shaft carries two pinions geering with two racks of similar pitch bolted to the under side of the travelling table. These racks are so placed that each tooth of the one shall be opposite to each space of the other; in this manner, the irregularity of motion so much complained of in ordinary rack-worked machines, as producing a waved surface on the work, is to some extent avoided. A more effectual method of attaining this end has been introduced by Mr. Collier of Manchester; this plan consists in making the teeth of the rack and pinion on what is technically termed the *step* system, that is, each tooth is divided in its breadth into three parts, each division being set a distance equal to one-third of the true pitch of the teeth, behind its neighbour. The practical result of this arrangement is, that although the strength of the original coarse pitch is preserved, yet the teeth work with the steadiness due to a pitch three times finer, or so many times less as the number of divisions of the teeth amounts to. This plan is now universally adopted in all rack machines, as it is simple, easy of application, and completely effectual.

This, although in our opinion not the best, is probably the most universally used species of driving geering applied to planing machines. Of the two remaining systems, the *chain* and *screw*, the latter, in point of mechanical elegance, is decidedly to be preferred. Mr. Whitworth's screw planing machine, which is worked in the same manner as his slotting machine, Plates XXXV., XXXVI., is perhaps the most finished specimen of modern tool making extant. The annexed cut, fig. 55, will serve to explain the method adopted by this engineer to give motion to the screw, as applied to his self-acting double cutting machines. Here *a* is the driving pulley for one direction of motion of the table; it is keyed upon the hollow shaft *b*, which works in a bearing *c* attached to the end of the bed of the machine. The shaft *b* forms at the same time the boss of the meter wheel *d*, which is cast on its end; this wheel geers with another wheel *e*, exactly similar, keyed on the end of the screw shaft *f*, which passes right down the centre of the machine immediately below the table, to which it gives motion by means of an arrangement of antifriction pulleys (see Plate XXXVI.) The contrary motion of the table is obtained by the pulley *g*, which is keyed on a solid shaft *h*, passing down the hollow boss *b*, its extreme end being supported by a bearing *k* attached to the bed. This shaft again carries a third meter wheel *l* geering with the one on the screw shaft *f*. The centre pulley *m* is loose on the driving shaft, and serves to carry the belt when the machine is

Fig. 55.



inert. When the strap is shifted from this pulley to either of its neighbours on its right or left, it will be observed that the arrangement of the geering will cause a change of direction of motion of the table accordingly. As this motion gives the same speed in both directions, it is obviously inapplicable to machines which cut only in one direction, as it would involve a considerable loss of time in the return stroke, for which provision is always made in the ordinary machines.

The additional speed for the return stroke has been applied to similar geering by Mr. Shanks, of Johnstone, near Glasgow. The place of the central bevil wheel is, in this case, supplied by two wheels, whose diameters bear the same ratio to each other as that of the two required speeds for the back and forward motions. The arms of the larger wheel are hollowed inwards, for the purpose of bringing the pitch line of the teeth of the smaller wheel, which is bolted to it, into the same plane with that of its own; the two reversing pinions are then attached respectively to a hollow and solid shaft, each geering with its own wheel, and, of course, changing the speed of the table accordingly. The additional wheel renders the geering heavier, and a disagreeable rattling motion is produced when the quick speed is in geer; still, if properly constructed, we see no reason why it should be at all inferior to Mr. Whitworth's simple arrangement, which is considered in every respect satisfactory.

The principle of anti-friction rollers acted on by a screw, as a means of obtaining a rectilinear motion, was first introduced by Mr. Whitworth in 1835, when he employed it as a motion for the carriage of the self-acting spinning mule. In his planing machine, the rollers are placed parallel, face to face, on opposite sides of the screw, their axes revolving in bearings attached to the under surface of the bed, and their peripheries projecting into the spaces between the threads of the driving screw. It will be seen that each periphery has two opposite points of contact, acting alternately, according to the direction of motion of the screw, which, as it revolves, brings its threads to bear upon the rollers, causing them to revolve, and, at the same time, to carry forward the table to which they are fixed. The friction which would occur, if the threads of the screw bore simply against a fixed nut, is thus transferred to the axes of the rollers, where the velocity is reduced in the proportion existing between their peripheries and the circumference of their axes. The proportion which Mr. Whitworth has found to answer best for this arrangement is as 7 to 1. The advantage which this mode of driving has over the common rack and the chain will be perceived at a glance, as not only is the motion rendered perfectly uniform, a condition essentially necessary to the proper action of the cutter in producing a good surface, but the construction of the driving geering is rendered to the last degree simple.

The arrangement of catches employed by Mr. Mylne is good, but the geering connecting the catch shaft with the strap fork is capable of much simplification.

Referring to Plates XXXIX. and XL., it will be seen that there are two adjustable catches *n*,*n*, set in a groove running along the side of the table; these catches are not set in the same plane, but one projects out beyond the other in order to suit the levers *g*, which are cast to a tubular shaft working loose on the driving shaft. When the table is moving forward, one of the catches comes in contact with its lever, and turns it over, as seen in the end view of the machine. The boss of these levers again carries a third lever connected to the weighted lever on the end of the bed, which again communicates by a shaft running along the side of the machine, with the strap fork shaft C; the latter thus causes the shift of the straps from one pulley to the other, and reverses the motion of the table. Upon the return of the table, the catch which has just acted, now returns without coming in contact with its lever, as its former motion has placed it out of its reach, having at the same time raised the second lever on the same shaft, to an upright position, so that it may be acted upon by the other catch at the contrary end of the table, when the straps are brought back to their primary positions.

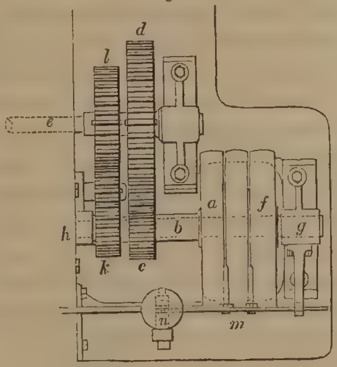
Another arrangement which is equally adapted for machines upon the chain, rack and pinion, or screw principle, has been applied with success by the writer, to various single acting machines. The arrangement of the driving pulleys is similar to that shown in fig. 55, the loose pulley being in the centre; one of the exterior pulleys is keyed directly on the pinion shaft, and gives the forward motion to the table. The other pulley is toothed upon its inner surface, so as to form an internal wheel, which is employed to drive a small pinion working upon a separate stud, and geering with another pinion keyed on the rack pinion shaft. In this way when the strap is shifted on to the latter pulley, it communicates a reverse motion to the table at a speed so much greater than that of the forward motion, as the ratio existing between the diameters of the internal toothed wheel and the connecting pinion upon the driving shaft. This motion is extremely compact, as it produces the increased return speed without requiring any additional pulleys or complicated mechanism.

As a driving and reversing geer for small planing machines, Mr. Nasmyth has applied the mangle wheel motion, so called from its adaptation as a continuous forward motion for common clothes mangles. It consists of a large disc, having near its circumference a circle of pins bolted through the metal at right angles to its plane; these pins answer as a set of teeth, into which a small driving pinion geers, working alternately on the outside and inside of the teeth so as to effect the desired reverse motions. In Mr. Nasmyth's arrangement, the driving pulley is keyed upon a light shaft, passing transversely beneath the table of the machine. The contrary extremity of this shaft, which projects beyond the edge of the bed, carries the mangle pinion geering with the pins of the mangle wheel. The latter is keyed upon a central transverse shaft which passes beneath the table of the machine and carries a large chain pulley. Round this pulley a chain is passed twice, and its two extremities are passed round two fixed pulleys placed at contrary ends of the bed, and attached to the opposite ends of the travelling table. The reversing of the mangle wheel, and consequently that of the table, is effected in the following manner: at two points in the circumference of the mangle wheel, one or two of the pin teeth are removed, and a sloping guide or stud is placed at each point, so that when the driving pinion arrives there, this guide causes it to traverse in or out, as the case may be, to geer with the inner or outer sides of the pins, under which conditions it is easy to see that the two contrary motions of the wheel will be the result. The guide supporting the pinion shaft is slotted horizontally to allow of the traversing of the shaft as well as to prevent its running beyond the point of geer with the pins of the mangle wheel, the bearing on the opposite end of the shaft next the driving pulley being arranged to swivel on a centre, so as to permit of this motion. This movement of the pinion shaft is also taken advantage of in giving the feed motion to the cross-slide of the machine, being connected to the vertical rod carrying the catches for the ratchet wheel of the transverse screw.

This movement, although ingenious, is destitute of the advantage of an increased speed in the return stroke, consequently much time is lost by it when applied to single-acting machines.

The arrangement applied by Messrs. Nasmyth and Gaskell, to their rack-planing machines, is a very convenient, though somewhat cumbrous motion. Fig. 56 is a ground plan of this geering, in which *a* is the forward motion driving pulley, keyed on the hollow shaft *b*, which carries a pinion *c* geering with a large spur wheel *d*. The latter is keyed directly on the rack pinion shaft *e*, shown in dotted lines passing beneath the table of the machine. The backward motion pulley *f* is keyed on the solid shaft, passing through the hollow one and revolving at one extremity in the bearing *g* fixed on a pedestal attached to the bed plate, and at the other in the bearing *h* bolted to the side of the bed. This latter shaft carries another pinion *k* geering by means of an intermediate carrier wheel, with the spur wheel *l* also keyed on the rack-pinion shaft. The centre pulley is of course loose, serving merely to carry the strap when the machine is stopped, and during the transfer from the forward to the backward pulley. Thus it will be seen that the return stroke of the table will be so much quicker than the cutting one, as the difference in diameter of the two wheels *l* and *d*, or rather, as the ratio which exists between the wheels *k* and *l*, and *c* and *d*.

Fig. 56.



The strap fork is seen at *m;* it is worked by catches fixed on the other side of the table, a connecting shaft from which passes beneath the bed where it is attached to the fork; *n* is a weighted lever for the purpose of giving a sudden shift to the strap, so as to give the workman a better command over his machine.

Of chain-worked planing machines, the modification introduced by M. Decoster, of Paris, is perhaps one of the most complete. In his machine, he has made use of the driving geer shown in Fig. 55, as applied by Mr. Whitworth to his screw-machines. In the particular example by M. Decoster, to which we refer, the chain motion is applied to give motion to the tool slide, while the table of the machine remains stationary. This plan is found extremely useful in planing heavy and unmanageable pieces of metal, as the latter may be firmly secured to a foundation independent of the machine, while the tool alone traverses over it; and, consequently, no more power is absorbed by a heavy casting, than by the lightest possible piece of metal. The driving geering before referred to, is here placed alongside the bed of the machine, near one end; the pinion on the central bevil wheel geers with a large spur wheel, on a shaft passing transversely across the bed of the machine, below the table. The latter shaft carries two rag wheels, placed near its two extremities just within the frame of the machine. Round each of these wheels is passed an endless chain, which passes along the whole length of the machine, returning round a similar pair of wheels revolving loosely on studs at the contrary end of the bed. The upper length of this chain is attached to the lower surface of a V-grooved slide, working in corresponding grooves planed in the upper surface of the bed. This slide carries a second horizontal slide supporting the tool holder in the usual manner. The feed motion of the cross-slide is ingeniously effected by two ratchet catches attached to the spur geering on the end of the horizontal screw. The lower extremities of these catches are set to come in contact with moveable tappets attached to the fixed frame of the machine, so as to give the proper amount of motion to the screw of the cross-slide. The method of attachment of the driving chains adopted by M. Decoster has the advantage of giving a steadier pull to the tool-slide, than can be obtained by the central mode of fastening, with a single chain.

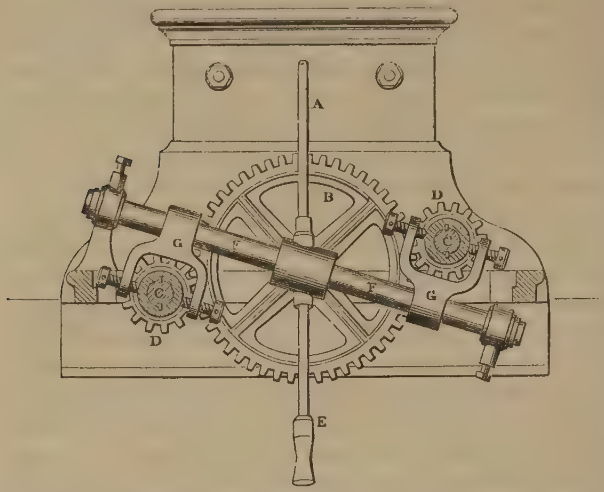
The principle of the moveable tool and fixed table has also been adopted by M. Cavè and Mr. Hick of Bolton. In M. Cavè's machine, the driving motion is given to the tool-slide by an endless strap. The driving pulley is placed immediately over the centre of the bed of the machine, the strap from which passes below two fixed tension pulleys, placed just beneath the driver, and thence round two fixed pulleys attached to the opposite ends of the bed. The attachment of this strap to the sliding frame of the tool is effected by passing the strap in contrary directions round two separate pulleys, each carrying a pinion geering with a central driving wheel. The shaft of the latter passes across the bed of the machine, and carries two pinions, geering with two racks, placed within the framing, and running along the whole length of the bed.

The arrangement of the spur reversing geer will be understood by referring to fig. 57 which is a side view of the tool slide, frame and geering, with the driving pulleys removed. A, is the travelling tool slide, carrying the central driving wheel B, keyed on the pinion shaft—the shafts C, C each carry a loose driving pulley, capable of connection by means of sliding clutch-boxes with the two pinions D, D. These latter work loose on the pulley shafts, and geer with the central wheel B, so as to drive it in either direction accordingly as the clutch-boxes are set.

Two moveable inclined tappets are fixed to the bed of the machine, which alternately come in contact with the lever E on the oblique shaft F, so as to move it in and out according to the motion of the slide. The shaft F carries the two forks G connected to the clutch-boxes of the pinions D, D;

which are placed, one on each side of it, so that when the lever E is pressed upon by its tappets, the hold of the two clutches is changed accordingly; one being thrown out of geer at the same time the other is put in. In this manner, as the two pulleys on the pinion shafts revolve in different directions, a reciprocating motion is given to the travelling slide.

Fig. 57.



The cross-slide of this machine is provided with two tool holders, one on each side, so as to cut in both directions; this improvement effects a great saving of time, as the return stroke is rendered equally as effective as the forward one.

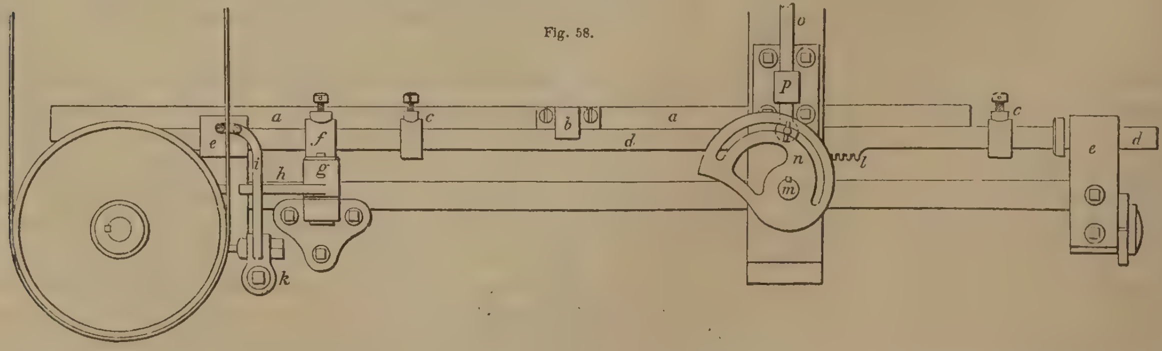
In Mr. Hick's moveable tool slide machine, the traversing motion is given to it by means of steel belts. The driving pulley of the machine is alternately worked by a cross and open strap; the shaft of this pulley is connected by means of spur geering, with a transverse shaft carrying two pulleys working outside the frame of the machine. These pulleys each carry an endless steel belt, running alongside the frame and passing round two similar pulleys placed at the contrary extremity of it. The steel belts are attached to projecting levers on the cross-slide by means of tightening screws, so as to communicate an alternate motion to it, accordingly as the open or crossed strap is working on the driving pulley.

As the speed of the travelling table of this machine is the same in each direction, it is arranged to cut both ways, by the adaptation of Mr. Whitworth's revolving Tool-holder—subsequently described.

It is a matter of considerable importance in planing machines, to have a compact arrangement at command, both for reversing the motion of the table and also for giving the self-acting feed motion to the cross-slide. As regards the reversing motion, in planing delicate or complicated work, it is often requisite to be able to stop the motion of the table within the shortest possible limits, as for instance, in planing up to an abrupt shoulder; in such a case, if the tool does not proceed sufficiently far, a surface of metal is left which must be removed by some more laborious means, or, on the other hand, if it proceeds a little too far, the tool strikes against the obstacle and causes an injury either to the work or to its own geering. In small machines, this is easily avoided, by the use of the crank or grooved disc, which allows of the greatest exactitude in the length of travel, but in machines of the larger class we are driven to some other expedient to attain this end. Where the change of motion is effected by the pulley belt, the simplest and most effective system of quick stoppage, is the addition of the weighted balance lever attached to the strap fork; the sudden fall, in either direction, of this weight causes an instantaneous motion of the strap and stops the table within very short limits Where still greater nicety is required, possibly the addition of a moveable clutch-box may be of some assistance. Some makers, indeed, have applied the clutch-box to the geering shown in fig. 55 instead of the shifting strap, the clutch being arranged to throw the two side bevil wheels in geer alternately with the centre one.

As a compact and efficient self-acting reversing and feed motion, we give that adopted by Mr. Whitworth, as one of the best. Fig. 58 is a side elevation of the apparatus; *a* is the table of the planing machine, on the side of which, at the centre, is screwed the fixed catch *b*, which in the course of working, alternately comes in contact with the moveable catches *c*, *c*, adjustable on the shaft *d* which runs alongside the table, sliding in the bearings *e*, *e*, at each end of the frame. This shaft carries a third adjustable catch *f*, connected with a short lever cast on the boss *g*, working loose on a stud screwed to the frame. The same boss has also cast upon it a second lever *h*, at right angles to the former one, the end of which works in a slot in the strap fork *i*. The latter oscillates on a centre attached to the bed at *k*; when the catch *b* comes in contact with one or other of the studs *c*, the shaft *d* is carried along laterally, and gives motion through the arrangement of levers just described, to the strap for

Fig. 58.

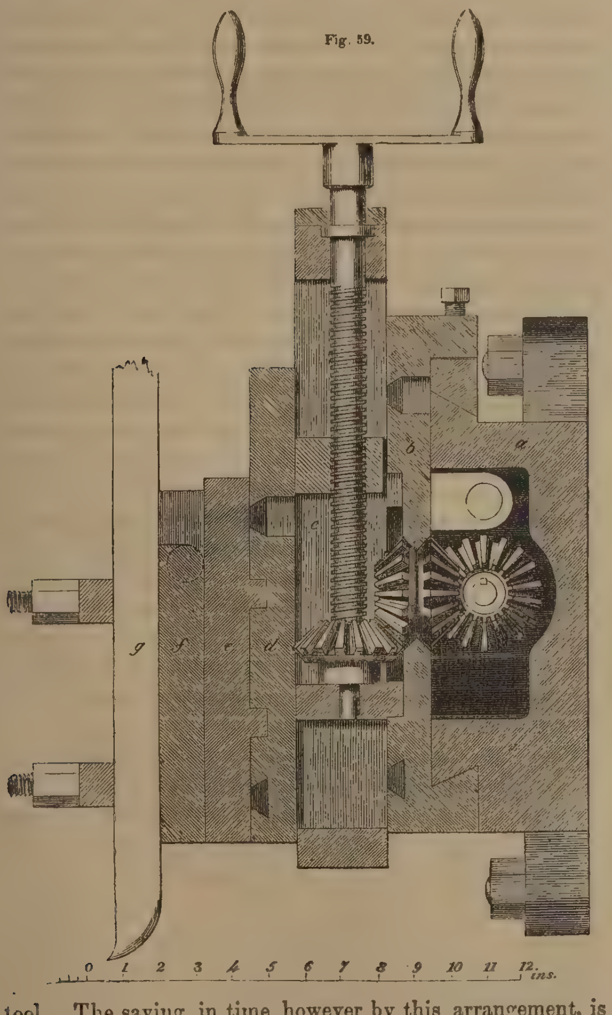


so as to shift the strap from one pulley to the other, and reverse the table.

The self-acting feed motion of the cross-slide is effected in the following simple manner: on the sliding shaft or rod *d*, a few rack teeth *l* are cut, which geer with a segment of a spur wheel, keyed on the shaft *m*, working in bearings screwed to the upright frame of the machine, and carrying the eccentrically grooved disc *n*, which revolves with it; *o* is the vertical rod carrying the ratchet catches for working the horizontal screw of the cross-slide; it is guided by a bearing *p*, screwed to the frame, and carries at its lower extremity a pin working in the eccentric slot of the disc *n*. Thus when the rod *d* receives its motion from the catch *b*, at the termination of the stroke of the table, its short rack causes the disc *n* to make a portion of a revolution, so as to raise or depress the rod *o* by means of the eccentric groove. This motion is at once effectual and easy of application, beside possessing that great desideratum in all tools—compactness.

We now come to the consideration of tool-holders. The specimen of a tool-holder given in Mr. Mylne's machine—is one of the more complicated variety; being provided with a double set of slides and appropriate screws, for the purpose of planing at two different angles with one adjustment of the tool. The saving in time however by this arrangement, is more than counter-balanced by the increased cost of the tool box, and the disadvantage which it entails upon the machine, by throwing the point of resistance in cutting, so far from the surface of the supporting frame as to render the cutting action unsteady.

Fig. 59.



A somewhat simpler modification of the same variety of holder, is represented in fig. 59, where the self-acting down cut motion is obtained by one screw; *a* is a transverse section through the cross-slide of the machine, to which is fitted, by dove-tails, the horizontal sliding plate *b*. The latter again carries the down-cut slide *c*, being attached to it by dove-tail headed bolts working in a circular groove in the former. The slide *c* is fitted with a central screw carrying a nut fixed on the front sliding plate *d*, so that the latter may be moved at any angle to the bed of the machine according to the angular position of the screw; *e* is a front plate, checked into the slide *d*, to which is hinged the tool-holder *f*, carrying the tool as shown at *g*.

The self-acting feed motion is given to the screw in the plate *c*, by bevil geering, similar to that employed in Mr. Mylne's machine. The hinge at the upper end of the tool-holder, is for the purpose of allowing the tool to give way in case it comes in contact with any obstacle during the return stroke of the machine. In all properly constructed tool boxes, the mechanism is arranged to lift the tool out of the way at each return stroke, so that it never rests upon the surface of the work in the back motion. This is effected by a separate transverse screw placed parallel to the main traversing screw, and worked by the same geering. In Mr. Bodmer's tool boxes this screw carries a nut with a slotted projection fitting to a pin in the upper end of the front plate, which oscillates loosely on a fixed centre. The nut being carried along with the tool-slide by the revolution of its screw, remains always immediately above the centre of the tool-holder; at the termination of a stroke, the reversing geering is so connected with the screw, as to give the latter a lateral sliding motion, the nut upon it then moves the front plate by the pin in its upper side. This plate carries a small inclined pin which in its motion presses against the front of the hinged tool-holder, thus raising it out of connection with the work.

The self-feeding down-cut motion is also given by the same oscillating plate. The latter is provided with a toothed sector screwed to it near its lower extremity, and geering with a small bevil pinion on the down-cut screw spindle. The latter being fitted with a ratchet wheel, receives at each stroke of the machine, an amount of motion proportioned to the material to be cut. This is probably one of the most complete and effective of all single acting tool-holders, and is a good specimen of the high degree of eminence which Mr. Bodmer has attained as a maker of constructive machinery.

Figs. 60 and 61 represent an external elevation and vertical section of the patent revolving tool-holder of Mr. Whitworth, by the use of which, the whole of the time at present lost in returning the tool for a fresh stroke is saved by causing the tool to cut in both directions. The tool is placed in a vertical cylinder A, which has a spiral groove along its surface, for the purpose of receiving a projection from the spindle B. The latter moves vertically in bearings attached to the front slide, and carries at its upper end, a box C, linked to the double lever D. E, E, are two adjustable studs, screwed to the vertical reversing rod O, (see fig. 58). At the termination of each stroke, the lever D receives motion alternately from the studs E, E, by which means the sliding rod B is moved vertically; the pin of the latter, which fits into the spiral in the tool cylinder, causes it to make at each movement a semi-revolution, so as to set the tool ready for cutting in the contrary direction.

Fig. 60.

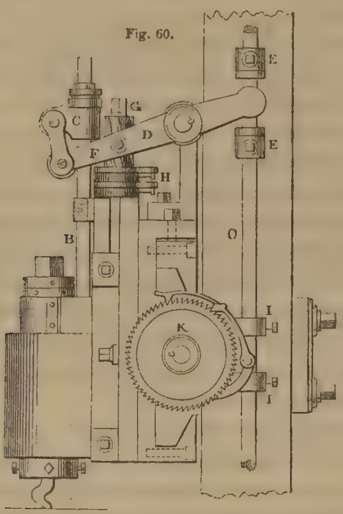
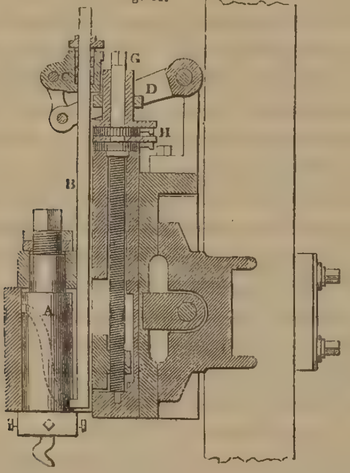


Fig. 61.



The self-acting down-cut motion is also given by the same arrangement of levers. The down motion screw is provided with a stud at its upper extremity, having a number of spiral grooves cut upon it, into which a series of pins projecting inwards from a ring jointed to the slotted lever at H, are arranged to work. It will be readily seen that the oscillating motion of the lever thus at the same time causes the down cut screw to make a portion of a revolution in a manner precisely similar to the action of the pin in the groove of the cylindrical tool-holder; a set of ratchets are fixed at H, for the purpose of allowing of the return motion of the spiral stud, without at the same time bringing back the screw itself.

The horizontal traversing motion of the tool-holder is effected by another pair of studs I, I, screwed on the same upright shaft O. These studs give motion to the catches of the ratchet wheel K on the extremity of the traversing screw spindle. Another modification of the revolving tool-holder has been applied by Mr. Whitworth, with considerable success. The reversing motion is, in this case, communicated by a set of band pulleys instead of the arrangement of levers in figs. 60 and 61. The place of the eccentric disc in that figure is supplied by a light wheel, which is provided with two adjustable pins, fixed in suitable radiating slots diametrically opposite to each other on each side the centre. To one of these pins the end of a light cord is attached, which thence passes round a pulley on the horizontal screw shaft of the cross-slide, to which it thus gives motion. From this pulley it proceeds to a second fixed pulley on the top of the upright frame whence it descends, and by means of a pair of guide pulleys, placed at opposite extremities of the cross slide, it is passed once round the grooved top of the cylindrical tool-holder, and is finally attached to the second pin in the slotted disc. Thus, at each movement of the latter, it will be understood that a semi-rotatory movement is given to the tool, the precise position of which, in a direct line with its traverse, is obtained by means of a fixed stop on the down slide, with which a projecting piece on the socket pulley of the tool comes in contact. Although this arrangement exhibits a good deal of ingenuity in its details, yet it is perhaps inferior to the plan shown in our last illustration, both as a specimen of elegant mechanism, and in the correctness of its action. The prominent appearance of straps and cords in conjunction with highly polished metallic surfaces, presents at once an incongruous appearance, and suggests an idea of a half-digested scheme, worked out only with the readiest materials.

In addition to the common rectilineal planing machine, machinists have of late years found a powerful auxiliary in the circular machine; this may be defined in general terms, as a lathe with a vertical spindle. The tool is either fixed or moveable, the former being the preferable and more general arrangement. The advantage which these machines possess over common turning lathes, are firstly, the greater facility of adjustment of heavy castings preparatory to planing them; and, secondly, the greater latitude they allow for acting on masses of metal of great diameter, as the driving wheels of locomotive engines, fly wheels, &c. Of this species of tools perhaps Mr. Bodmer's modification stands highest in the scale of usefulness. It consists of a heavy foundation plate, in the centre of which a strong vertical spindle revolves, having a horizontal circular table of large diameter keyed upon it, and provided in the usual manner with slots for fixing the work. The fixed cutter is held in a tool-box precisely similar to that adopted in the common planing machines. It is placed on a strong horizontal cross slide, which is adjusted to work freely in a vertical direction upon two upright frames, placed one on each side of the revolving table. The tool-holder being fitted with a down cut motion, is readily adjusted with great nicety to suit the work; besides which, its horizontal motion on the cross slide, combined with the vertical motion of the latter upon the uprights of the frame, permit the tool to be set to any portion of the radius of the table. The machine is fitted with a self-feeding motion—this is found very serviceable in turning up the tires of locomotive and carriage wheels, the rims of small fly wheels, &c. Messrs. Carmichael of Dundee have constructed a machine which combines the advantages both of the rectilineal and circular machines, with a view to the finishing of more complicated work than can be effected by either of these separately. The machine is provided with a rectilineal sliding table, as ordinarily used, supported on a fixed bed. A horizontal shaft passes transversely below the bed, and carries a plain sector of considerable radius, to which a chain is attached, and is connected to the opposite ends of the table. Motion is given to this sector by a plain grooved disc as usually applied to slotting machines; this is driven by over-head geering, supported by the upright framing which springs from the bed of the machine. The pin of this disc is attached by a suitable connecting rod to a crank keyed on the extremity of the shaft of the sector, which thus communicates a reciprocating motion to the table, of a length dependent on the position of the pin in the radius of the disc.

When continued circular work is required, the driving disc is thrown out of geer, and the driving shaft is connected with an upright spindle revolving in bearings attached to the upper framing of the machine, and carrying an adjustable cutter in a projecting arm at its lower extremity. This cutter is provided with suitable slides, by which it may be set at any required distance from the centre of motion of the spindle, so as to act upon any given circle. When thus arranged, it may be used as a powerful boring machine, by detaching the facing cutters, and fitting the proper tools as applied to the usual boring bars. When the combination of the straight and circular movements is required, as for finishing the flat sides and circular ends of strap links, &c., both movements are effected by the same driving shaft as follows:—On the end of the driving shaft are keyed two toothed sectors, arranged to give alternately a semi-revolution to the wheel on the driving disc for the rectilineal motion, and that on the train for giving the circular motion to the cutter spindle—each wheel being alternately held by a detent while the other is being driven. In this manner various kinds of work may be finished in a very superior manner, such as the plane and circular surfaces of plummer-blocks, connecting rod straps, &c.

In addition to these two descriptions of machines for obtaining a *plane* surface, a third species for planing curves has latterly filled an important office in the engineer's workshop. The tool to which we refer, is the compound machine by Messrs. Nasmyth and Gaskell, (see Plate LV.). This elegant little machine forms another link in the catalogue of automatic contrivances for superseding the tedious manipulations of file labour. By its assistance the circular ends of levers, connecting links, &c., are correctly cut out, and finished with a degree of celerity that sets manual labour at defiance. The essential principle of the machine is identical with that of a slotting machine with a horizontal movement, the tool slide being worked in a similar manner by a circular slotted disc. In planing the circular ends of levers, &c., the work, after being drilled, is fixed on a horizontal adjustable mandrel, with its rectilineal surface in a line with the direction of the traverse of the tool. A self-feeding motion causes the work to revolve slowly in the action of cutting, similarly to the same arrangement in the slotting machine. By detaching this geering, the tool becomes available for the production of plane surfaces at any angle, by an appropriate adjustment of the tool-holder—thus it unites the offices usually consigned to separate tools, and is a very useful auxiliary to the engineer.

Though not in immediate connection with the subject of planing, we may here mention Mr. Bodmer's *stand-cutting* machine. This tool is a species of planing machine, provided with a revolving cutter, and is used for the purpose of cutting out the recesses in the small stand bearings, &c., in cotton and other machinery, a species of work which requires the utmost precision and exactitude of management. In preparing and fitting up the supporting pedestals used for the rollers of spinning machinery it is essentially necessary to preserve their line of bearing perfectly level, otherwise the rollers will undergo an injurious strain in the working. To accomplish this in a speedy manner, Mr. Bodmer fixes a row of pedestals upright on a moveable bed, constructed like an ordinary planing machine. The upright framing in the centre of the machine carries a revolving cutter, similar to the steel cutters used for cutting the teeth of wheels; the row of pedestals, which are placed in a line with the motion of the bed, are then passed slowly beneath the cutter, thus securing an accurate adjustment of the height of each.

A machine similar in principle is used for fluting the wooden rollers of flax machinery, mechanism being introduced for the purpose of causing the rollers to make a portion of a revolution after the cutting of each groove.

Slotting machines, in their general principle of action, may be defined as planing machines with moveable tools. The period of their introduction to the workshop dates among the latest of the automatic tools of the day, as, until a short time back, the species of work now executed by them was entirely performed by the file and chipping tool. As a finished specimen of this tool, we may refer the reader to Messrs. Caird & Co.'s machine, Plates LXIV., LXV.

The method of transmitting motion from the driving geer to the reciprocating tool is here very simple, and the machine, as a whole, has a very handsome appearance. The table, in addition to the usual rectilineal and circular motions, is provided with apparatus for setting it at any angle, to suit the different varieties of work. This angular motion is useful in cutting the key seats of wheels, where a slight inclination is necessary to suit the shape of the fixing key

In such a machine as the one before us, it is evident that the size of the work, capable of being operated upon by it, is circumscribed by the distance from the cutting centre to the edge of the supporting pillar. If this distance is increased in order to suit the dimensions of wheels and castings of a large size, a greater disadvantage ensues, namely, an increased amount of unsteadiness of action. Messrs. Nasmyth & Gaskell have remedied this disadvantage most efficiently, by doing away with the supporting framing of the machine, and causing the tool to cut from below upwards. The machine is, as it were, entirely reversed in this modification, the driving disc and geering connected with the slotting bar being placed under ground, in a pit made for the purpose. The cutting end of the slotting bar projects upwards, through a fixed cast iron table on the floor of the workshop, upon which the work is laid in the act of slotting, As in this arrangement there are no supports to interfere with the work on the table, it is evident that this tool possesses an unlimited range of action. In another modification by the same firm, this object is attained by supporting the tool slide in a bottom plate attached to the floor, to which are attached four strong pillars, carrying a square table for the support of the work, at the height required for the convenience of the workman. The driving geer, in this instance, consists of a horizontal shaft, passing under the table, near the level of the floor, driven by a strap, and carrying a pinion geering with a large spur wheel, which, at the same time, serves to communicate the reciprocating motion to the slotting bar, being grooved across one side, for the purpose of receiving the traversing pin of the connecting rod. The self-acting motion of the table is extremely neat and convenient. The spur wheel shaft carries a small eccentric, the rod of which communicates with the ratchet wheels of two shafts running horizontally at right angles to each other, beneath the moveable slides of the table. Each of these shafts carries screws, one of which, passing beneath the centre of the table, gives the rectilineal motion, while the other geers with the screw teeth cut round the circumference of it, and, consequently, traverses the table in a circular direction.

In Mr. Whitworth's ingenious screw slotting machine, pretty nearly the same advantage, as regards an extension of space for the work, is obtained, without altering the original system of upright framing. Instead, however, of one pillar placed behind the table, he has introduced two, one being fixed on each side of the machine, the slotting bar being supported and guided by cross beams, running between the two, as distinctly shown in Plates XXXV., XXXVI. Here the slotting bar consists of a heavy cast iron beam, which carries near its centre the anti-friction pulleys before referred to. The vertical screw geering with these pulleys receives its motion from the bevil geer at the top of the machine. The self-acting motions of the table are very complete, and embody all the latest applied improvements connected with simple tools. To avoid the necessity of setting forward the work at each cut of the tool in the act of slotting small work, Mr. Whitworth uses a slotting bar provided with several cutters, each set a little behind its neighbour, so that one cut finishes the slot to its full depth. This arrangement is found to be very expeditious in light work, and ensures a regular depth of cut, when applied to the slotting out a great number of similar articles.

The reciprocating geering of the slotting bar, considered with reference to the ordinary machines, strikes us as being somewhat more complex than the nature of the machine requires. The round-about mode of changing the direction of motion of the tool, by the shifting strap and bevil geering suffers considerably in comparison with the simplicity of the plain grooved disc. It would also appear that some time must be unavoidably lost at each change of the strap, by the slip which must occur from the impetus of the pulley and geering, tending to carry it forward in a contrary direction to the motion of the strap. This objection, it is true, may be in some degree diminished, by setting the reversing motion a little in advance of the point at which the bar should return, so as to reverse it in time. Of the different constructions of upright framing, Mr. Bodmer's semi-circular frame, is eminently deserving of notice. This framing consists of a single casting, semi-circular in its horizontal section so as to suit the contour of the table of the machine. The top of this frame, which is also curved forward so as to stand over the centre of the table, is provided with a deep eye as a bearing for the shaft of the ordinary traversing disc as usually applied to machines of this class—the driving geer being placed behind the frame. This frame combines the advantage of great strength with little weight of metal, and its rounded form brings the whole strength of the metal to bear directly over the resisting point of the tool.

M. Cavè of Paris has gone a step in advance of the English engineers in the construction of this tool. He has endeavoured to combine in one single machine, the separate offices of slotting, boring, turning, and circular planing, and, considering the variety of motions required in such an arrangement, a very creditable machine has been the result. The frame of the machine consists of four cast iron fluted pilasters bolted together by a series of horizontal transverse pieces, so as to form a strong rectangular frame. The spindle of the machine is hollow, and is worked when in the act of slotting, by two grooved discs, placed one on each side of it, on two horizontal shafts connected to the driving geer. The pins of these discs are connected to a horizontal cross-head attached to the upper end of the spindle which thus receives a reciprocating motion from the revolution of the discs The parallelism of the spindle and guides is preserved by additional slides attached to the inside of the frame, in which a set of guide blocks on the cross-head work. When adapted for boring or turning, the driving discs are disconnected from the spindle, and a long hollow pinion connected to the driving shaft is brought into geer with a large spur wheel keyed on the spindle. The downward motion for the feed is effected in a peculiarly ingenious manner. On the top of the spindle, a cast iron socket is fitted and retained in its place by two pins passing on each side of the central feed screw. This socket has on its upper side a grooved circle in which the stud of a spur wheel may be set at any point. The same stud also carries a small spur pinion geering with a larger wheel keyed on a nut upon the down motion screw which works inside the hollow spindle. In this manner, a downward feed motion is given to the spindle in direct proportion to its angular speed on its own axes. In order to adapt it for circular planing, a horizontal slide of a radius equal to that of the table of the machine is attached to the lower extremity of the spindle; this slide is provided with a screw and tool-holder by which the cutter may be set to plane at any radius.

Mr. Stanfield of Bolton has lately introduced a clever arrangement of self-acting feed for a slotting machine. The shaft of the grooved driving disc carries a second eccentrically grooved disc in which a pin attached to a horizontal oscillating double lever works. The two ends of this lever each carry a connecting rod passing down each side of the machine, when they may be connected at pleasure with any of the usual rectilineal or circular motions of the table. Well finished machines of this class, when provided with a rectilineal and circular motion of the table, may be made extensively useful in cutting out any regular curves in metal, by working both motions at once; indeed new applications of this now indispensable machine are daily arising which render it an important object in the list of re-productive tools.

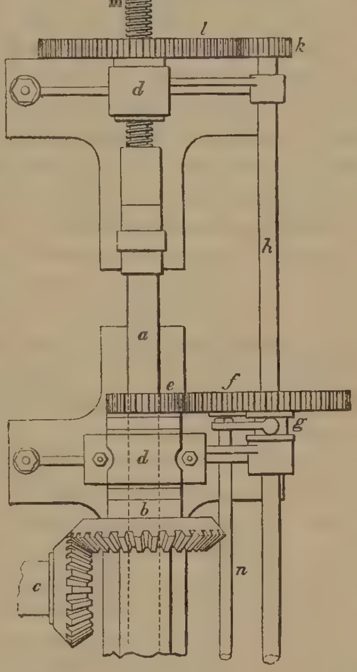
We have already discussed the philosophy of the true form of cutting edge for drills, as well as the minor species of tools of this class, (see ante pp. 159—171); it remains for us therefore now to enter upon the construction of what are more properly termed drilling *machines.* The varieties of these useful machines, as used by the engineer, are so numerous, that we can only find space to touch upon a few of those best adapted to the wants of the workshop. Practically speaking, drilling machines are divisible into two classes only, namely, the common vertical pillar or wall-side drill, and the radial machines.

For a good example of the former of these varieties, we may refer to the detailed views in Plates XXXIII., XXXIV. of Mr. Whitworth's vertical drill. This machine, which probably takes the first rank in its class, is independent, being provided with its own separate frame intended to be screwed to the floor without the additional support of a pillar or wall. Motion is communicated to the drill spindle by an arrangement similar to the back geer of a lathe, contained in an opening in the upper portion of the frame. The rectilineal feed motion of the drill spindle is self-acting, it is a beautifully ingenious arrangement, and is pre-eminently deserving of attention. The upper portion of the spindle between its bearings, is screwed for the purpose of geering with the inclined teeth of a pair of worm-wheels, placed one on each side of it; the axes of which work in bearings attached to the front of the frame. A compact friction-clip worked by a vertical screw from below embraces the projecting ends of these axes, by which arrangement the revolution of the wheels may be completely stopped when requisite. Thus, we will suppose the tightening screw of the friction-clip to be screwed up so that the latter holds the worm-wheels firmly in their position; it follows then, that if the drill is set in motion, the threads upon the spindle will act upon the teeth of the worm-wheels exactly as in a nut, and a quick descent of the spindle will be the result. If now the friction is slightly relaxed, then the speed of the descent of the spindle will have diminished so much as is due to the slipping round of the worm-wheels. In this manner any amount of feed motion may be communicated to the spindle with a nicety unattainable by any other means, so that the varying hardness of the metal under action may be immediately accommodated by a speed exactly suitable for cutting most advantageously. The table of the machine is provided most completely with all the requisite movements, and may be raised or lowered to suit the work by a pinion working in a vertical rack attached to the front of the frame.

In Plate XLIII. we have given complete views of Messrs. Nasmyth and Gaskell's drilling machine, which differs from the last specimen in the fact of its being destitute of a self-acting feed motion. The downward motion necessary for the feed of the tool is generated by the pressure of the foot acting on a treadle lever, which is connected by a vertical rod at the back of the frame, with a second lever working on a bearing at the top. The front end of this lever works a sliding bearing fitting on the top of the spindle, which thus receives a downward motion according to the pressure of the foot; the counter-weight on the back end of the lever bringing up the spindle again, on the lever being released from the pressure of the foot. This, as an independent variable motion, is very convenient, though inferior in nicety to that used by Mr. Whitworth.

Fig. 62 is a front view of the self-acting feed motion applied by Mr. Lewis of Manchester, to a wall side drill. Here the vertical spindle *a* carries a bevil wheel *b*, into which a second bevil wheel *c* on the driving cone shaft geers. The spindle works in two bearings *d*, *d* attached to a vertical plate bolted to the wall. A small spur-wheel *e* is keyed on the spindle, a little above the lower bearing; this wheel geers with a second wheel *f*, which carries a sliding bush *g*, so that it may be thrown in and out of geer with the driving wheel *e* at pleasure. The shaft *h* carrying this latter wheel passes up the side of the drill, and carries a second spur pinion *k*, at its upper extremity, geering with a large wheel *l*, the bush of which carries a nut working on the screw *m* on the spindle.

Fig. 62.



This motion may be driven by hand at pleasure by disconnecting the wheel *f*, and working a hand wheel fixed on the spindle *h*.

This may be taken as a specimen of the best kind of self-feeding motions, it is compact, effective, and inexpensive.

Of pillar drills, the modification introduced by Messrs. Randolph Elliot and Co. of Glasgow, is one of the most compact. This drill is certainly the simplest and most generally applicable of its class, its component parts being confined to a set of brackets to carry the drill spindle, without any additional frame, the office of which is supplied by any of the pillars of the workshop. The spindle brackets, as well as those for supporting the table, are clamp-bolted to the pillar—the spindle being worked either directly through a speed cone or by double geering, as applied to lathes generally. The self-acting feed motion in its general details is similar to that of Mr. Lewis, fig. 62. The table is provided with a strong central foot, to which two projecting brackets at the foot of the pillar are bolted; it is thus fixed immoveably, and so far lacks the important advantage possessed by a moveable table. In some varieties of drills, the feed motion is confined entirely to the table, which is worked by a rack and pinion movement, its weight being balanced by a counterpoise attached to a chain passing over a pulley on the top of the frame. In drills intended for boring to a considerable depth, this arrangement may have the advantage of steadiness of action; otherwise the moveable spindle is in every respect preferable.

In Plate XLIV. we have given two views of Mr. Whitworth's modification of the second class of drilling machines, the radial drill. A reference to this drawing will show that it is an arrangement on a principle quite distinct from that of the ordinary drilling machine. In the radial drill, the motion of the spindle is not confined to the vertical direction but it is provided with a lateral motion, whereby it may be set to any portion of the work within the limits of the radial arm, thus superseding one of the rectilineal motions of the table, which in this arrangement need only travel in one rectilineal direction beneath the drill. This machine is likewise provided with the same feed motion as applied by this maker to his ordinary vertical machine.

Messrs. Hick, to whom the merit of introducing this machine is due, make use of a somewhat different arrangement of driving and feed geer. The central frame of their machine consists of a short hollow circular bracket, having a nut fitted on its upper end, for the purpose of receiving a strong square-threaded screw, which carries the radial arm for the tool-slide. The vertical motion of the drill is therefore not confined to the motion of the spindle in its own bearings, but the whole radial arm is capable of a considerable vertical movement, by turning the nut on the central screw. The fast and loose pulleys for the driving geer are placed on a vertical shaft working in a bearing on the top of the radial arm immediately over the centre of the screw—motion being communicated from it to the drill spindle by means of bevil geer as in Mr. Whitworth's machine. The additional horizontal traversing motion of the spindle is effected by means of a pinion keyed on a hand-wheel shaft and geering into a rack on the radial arm; the downward motion for the feed is given by hand by means of an overhead ratchet lever connected to a ratchet wheel on a short horizontal shaft, over which a chain connected to a cross-head on the drill spindle is wound. A rod from this lever is brought within the reach of the workman, so that he can depress the spindle at pleasure, the upward motion being accomplished by means of a counter-weight attached to a chain pulley on the ratchet wheel shaft. A somewhat similar mode of central movement has been adopted by Mr. Bodmer in his improved radial drill. Here, the main central support consists of a strong hollow circular pillar provided with wrought iron centres at top and bottom, upon which it revolves. A screw passes down the centre of this pillar, and carries a nut attached to a bevil wheel on a bracket projecting through a vertical slot in the pillar. This bracket is screwed to the radial arm of the machine, and is raised or lowered by a lever handle on a shaft carrying a bevil wheel geering with the one referred to above as carrying the nut on the central screw. The driving geer consists of an overhead horizontal shaft carrying a bevil pinion working in a large bevil wheel on the top of the supporting pillar; the latter carries a spur wheel geering with a pinion on a vertical grooved shaft working in bearings attached to the pillar. This shaft carries a sliding bevil wheel provided with a key to fit its groove, so that it is at liberty to rise and fall with the radial arm without revolving loosely on its shaft. From this bevil wheel, motion is transmitted by a train of geering to the drill spindle in the radial arm, in the usual manner. The self-acting feed motion is highly ingenious and effective; the horizontal driving shaft of the radial arm, carries a small band-pulley from which motion is given to a short horizontal worm wheel shaft, the worm of which geers with a wheel on an upright shaft carrying a long pinion on its upper extremity. This pinion geers with a spur wheel attached to the nut of the top driving screw of the spindle, which thus receives a regular descending motion according to its boring speed. By detaching this geering, the long pinion may be worked by hand at pleasure. Although for some specific purposes this species of drill is highly useful, yet where great accuracy is required, it is inferior to the ordinary pillar or wall side machine, on account of its want of steadiness and rigidity.

Boring machines are, abstractedly, merely modifications of the larger class of drills, and are applied to the same purposes.

They are divisible into two classes, namely, horizontal and vertical machines. The latter species of machine is generally considered to be the most useful for engineering works, and there is no doubt that it possesses some great advantages over the former one. In the first place, the vertical position of the cylinder entails no transverse strain upon it, which would be pretty considerable, in a cylinder of large size placed horizontally. Such a strain would, of course, render the action of the tool extremely uncertain, and would detract materially from the required truth of surface. Again, the boring bar may in some sort be considered as liable to the same disadvantage, which its vertical position remedies. Lastly, the action of the cutters is not at all impeded by the presence of the turnings, which immediately fall to the foot of the cylinder and leave the cutter free at each progressive step. The most convenient and secure place for a boring machine of a large size, is the corner of the workshop, where the two angular walls form firm supports for the framing of the machine.

Messrs. Nasmyth and Gaskell have produced a useful tool of this particular class; the framing consists of a stout circular bottom casting, provided with a clamping ring for holding down the cylinder in the act of boring, and an overhead crossbeam for carrying the upper extremity of the boring bar spindle. The footstep of the boring bar is placed beneath the ground floor of the shop, where, as well as for a portion of the driving geering, an excavation is purposely made to receive them. The driving pulleys are placed outside this pit; they communicate with an oblique shaft, which carries a worm on its end, geering with a large worm wheel near the foot of the boring bar, to which motion is thus given. The cutter boss is traversed in the usual manner by a longitudinal screw in the bar, but a different method is adopted for returning it after the first rough cut. As soon as this is accomplished, the cutter boss is detached from the nut of the traversing screw, and is hauled up alone by a small crane attached to the machine, the cutters are then re-set and the finishing cut is gone over.

A machine somewhat similar, but of gigantic dimensions, has been constructed by the same engineers for the Great Western Steam Navigation Company, for the purpose of boring out the cylinders of the Great Britain steamer.

In this machine, the entablature carrying the upper end of the boring bar is supported on two massive pillars of masonry, placed one on each side of the boring bar. The feed motion of the cutters is novel and ingenious in the extreme; it consists, primarily, of an internal screwed collar fixed on the upper surface of the entablature, and surrounding the boring bar. A train of geering, terminating in a pinion working into a rack running down the side of the boring bar, is attached to the latter and revolves with it. The first wheel of the train is a species of crown wheel, its teeth being set at right angles to its axis of motion; this geers with the internal threads of the screwed collar before mentioned, so that by this means, the train is set in motion by the revolution of the bar, and the cutter boss, which is attached to the lower end of the rack, is raised and lowered at pleasure.

Mr. Walton of Leeds has introduced a highly effective boring machine, with columnar framing intended principally for boring the apertures in the tube plates of locomotive engines. The machine is capable of drilling a series of parallel holes on a surface of five feet square, without refixing the object under operations, the tool holder and the table being moveable at right angles to each other. This boring machine may be considered as a magnified drill, as the spindle is fed longitudinally, no cutter boss being attached. The framing consists of two plain columns, coupled at the top by a suitable entablature, and carrying two other transverse beams for the support of the drill spindle and driving geer. The self-feeding motion is similar to that illustrated by fig. 62, and it may also be worked by hand in the same way.

The spindle is capable of a vertical travel of 24 inches, and is consequently well suited for boring out the small cylinders of locomotives, &c., as well as for boring out the eyes of carriage and other wheels, which it will receive up to 6 feet in diameter.

Plates XLV. and XLVI. now introduce to us the second species of boring machine, the horizontal one. Our example is intended for the heaviest kind of work in the engine shop, and is purposely very strongly constructed. The arrangement of the train of driving geer allows of a considerable latitude of speed of the cutters; the changes by alteration in the geering permit of so small a variation as 1 to $1\frac{1}{4}$, so that any speed within the entire range may be obtained within $\frac{1}{8}$th of the one required. In Plates XLVII. and XLVIII. we have given a still more complete tool by Messrs. Kinmonds, Hutton and Steel, of Dundee. It is provided with a slide rest and tool holder, and thus becomes available for turning as well as boring. This is a very valuable machine, and where ground space is an object in the workshop, will be found highly convenient from the saving of space which it effects. The proper speed at which the cutter should pass over the surface of the metal may be stated as from 6 to 7 feet per minute, though this must, of course, be dependent in a considerable degree upon the relative hardness of the metal. Until a few years back, cannon were cast with a core, in the same manner as steam engine cylinders are now. Experience has, however, pointed out the fallacy of this method, as it was impossible to make a true bore, the cutter having a tendency to follow the inaccuracies of the yet rough surface. In addition to this, the guns, when cast hollow, have a tendency to become spongy near the inner surface of the bore. To remedy these defects, all ordnance are now cast solid, and thus the metal is rendered closer in the grain, and any "blown" or defective parts are confined to the centre of the metal, when they are cut out in boring. Guns are bored in a manner directly the reverse of that employed in preparing steam engine cylinders. The bed of the machine employed for boring out guns is provided with two pedestals having bearings of a considerable diameter, in which the gun itself is caused to revolve, the cutter remaining stationary on a sliding carriage. The cutter is kept up to its work by means of a weighted lever, attached to a pinion shaft beneath the bed of the machine. This pinion geers with a rack attached to the cutter carriage, which is thus impelled forward towards the gun, until the weights in the loaded lever reach near the ground, when the lever is raised and the weights re-set by a ratchet wheel.

The punching machine, from being confined to the tinsmith and ornamental metal worker's shop, has latterly become an instrument of no slight importance to the engineer, whose ponderous adaptations of this tool excite the greatest wonder in the mind of a spectator unaccustomed to their operations. Here the punch or cutting tool seems to pass through an enormously thick piece of cold metal, as if the latter was so much pasteboard, the whole of the operation being conducted without producing the least noise.

In Plate LI. we have given various views of a compact and convenient punching machine, constructed by Messrs. Caird of Greenock. The framing consists of one solid casting, open in the centre for the reception of the large driving bevil wheel on the horizontal punching shaft. The driving geering and fly-wheel for steadying the motion are placed overhead, so as to be completely out of the way, and leaving a clear space all round the machine for the workmen. The shearing and punching apparatus being on two opposite sides of the machine, these two operations may be conducted at the same time, without any risk of confusion amongst the men.

Of late years, the system of riveting by steam power has made rapid progress, and promises in all large works to supersede entirely the old laborious and noisy process of riveting by hand. We are indebted for the introduction of this machine to Mr. Fairbairn of Manchester, who invented it upon an emergency to supply the loss of a set of hand riveters who had left their employment on account of some business disagreement. Mr. Fairbairn's machine (see Plate LII.) is an elegant and efficient apparatus. The essential principle of the machine is that of the knee-joint lever, as applied to printing presses. Motion is given to this lever by a revolving cam which acts upon a loose anti-friction pulley, placed at the centre joint of the lever. The saving by the use of this machine is very great, and, what is of no small consequence, the whole operation of riveting is performed in silence. The system of pressure, too, has the advantage of leaving the metal of the rivets in its original state, whereas by the method of hammering, the rivets are subject to crystallization, and as a natural consequence, numbers of the heads give way in the finishing.

We have seen a section of two pieces of boiler plate riveted together in this manner, the section being taken through the centre of the rivet, and it was remarkable to see how the pressure of the machines had forced, as it were, the metal of the rivet into all the interstices of the plates, incorporating the two so as to be scarcely distinguishable from one piece of solid metal. A further improvement of the process has been effected by M. Lemaitre, of Paris, who has combined the two operations of riveting and punching, in one machine, which he actuates by the direct pressure of steam, without additional geering. We have detailed this machine fully in Plate LII., where also is shown M. Lemaitre's ingenious arrangement for riveting long and narrow tubes.

Messrs. Schneider of Creusot, in France, have likewise applied the direct action of the steam to the purpose of riveting. Their modification closely resembles in principle the plan adopted by Mr. Fairbairn; the piston of the steam cylinder is jointed directly to the central joint of the "knee" combination, and the rise and fall of the former gives a lateral traversing motion to the compressing die. Still more recently, Messrs. Garforth of Duckinfield have applied the pressure of steam even more directly. The bar which carries the compressing die, for forming the rivet, is at the same time the piston rod; the cylinder of which is placed horizontally in the frame of the machine. It is evident that in this plan, as no mechanism intervenes between the point of action of the steam, and the point of resistance, a very large cylinder must be used to produce the required pressure; but as the stroke of the piston is proportionally shortened, there is no larger amount of steam expended than in the small piston machines, and the friction is less.

Mr. Walton of Leeds has produced a very useful punching and shearing machine, capable of being worked either by manual or mechanical power. The framing consists of a single casting, having a stud keyed in the thickness of the metal near its top, to carry the fly-wheel and driving pinion, which are cast together. These work loose on the stud, the pinion geering with a large spur wheel keyed on the horizontal eccentric punching shaft. A slot is cast through the centre of the frame for the reception of this shaft, suitable bearings for carrying it being placed within the slot; the projecting end of the shaft is slightly eccentric, to give motion to the vertical punching and shearing shaft. The latter consists of a heavy piece of metal, having a horizontal slot in the centre, so as to allow a clear space for the lateral working of the eccentric end of the driving shaft. Bearings are attached to the front of the frame, in which the punching slide works; the top of the latter being the shearing end, and the bottom carrying the punch, the matrix for which is fixed in a projecting piece cast on the frame.

The machine is adapted to punch holes up to $\frac{3}{4}$ inch in diameter in plates $\frac{3}{8}$ inch in thickness, at any distance from the edge not exceeding $7\frac{1}{2}$ inches, the frame being hollowed out to this extent to permit of the entrance of the plates. The shears are capable of cutting plates $\frac{3}{8}$ inch in thickness, and 12 inches in breadth, without curling the piece sheared off.

The construction of this machine is exceedingly simple, and being set in an independent framing of its own, it may be moved to any part of the workshop with facility. A somewhat similar machine, but much more complete in its details, has been constructed by Messrs. Nasmyth and Gaskell. Here the punching slide is provided with four punches, by which means the same number of holes are punched at each stroke of the machine. The punching operation is also made self-acting, by an arrangement of a self-moving table for carrying the work. The plates intended to be punched are fixed in the usual manner on a travelling table, moving on wheels set to run on a pair of rails. A long notched bar is attached by brackets to the under side of this table, and is arranged to traverse the table in the following manner:—The large driving wheel on the eccentric shaft carries a pin fixed in the side of its rim, which, once during each revolution, comes in contact with a lever connected to a ratchet catch adapted to take into the notches of the bar; thus each revolution of the spur wheel causes the table to advance a distance equal to the pitch of the notches in the bar.

Plates LIII., LIV., represent a machine intended for the bending of wrought iron plates. This machine, owing to the increase of iron ship building, has latterly risen to be of great importance to the engineer and ship builder. The present machine being principally intended for the use of the ship building yard, where few plates are required to have a regular curve throughout, is not provided with geering for simultaneously altering the positions of the ends of the front roller. The arrangement adopted allows of the setting of one of the ends of the roller at any position with regard to the other, so as to give any required twist to the plate.

In the original application of the bending rollers to the curving of boiler plates, none of the rollers touch each other, and they are placed so that lines drawn from centre to centre form an equilateral triangle, the upper central roller being made adjustable for the different curvatures required; but this arrangement is entirely superseded by that adopted in Mr. Napier's machine.

ENGINEER AND MACHINIST'S ASSISTANT.

PART FOURTH.

ON THE

THEORY AND CONSTRUCTION OF WATER WHEELS.

Although in localities where mineral reservoirs of motive power are convenient, the ever-available steam engine has much diminished the importance of hydraulic movers, these must always continue to be the most economical, and therefore the most frequently resorted to in situations where the liquid element can be attained in sufficient abundance, and under the necessary circumstances to answer the conditions contemplated. A waterfall is a gratuitous bequest of nature, rendered available without labour, and furnishing its supplies without the intervention of human aid. The energies of the steam engine, on the contrary, can be commanded in any situation, only by the influence of the miner; and in localities much removed from sources of fuel, can only be sustained at an expense which falls heavily upon the operations to which they are subservient. That expense, it is true, is continually being diminished, and by means of the steam engine itself, in its character of a carrier; but no happy discovery—no possibility can reduce it to the minimum at which our water-runs are maintained.

But while water power has the advantage of economy, where it is abundant and constant, in other localities where it is more immediately dependent upon the condition of the seasons, it is under the disadvantage of being less certain, and less under control, than the more artificial agency developed in the steam engine. It is this independence of time and season, of circumstances and locality, which marks the great superiority of this potent creation of engineering skill, and which, in its multiform applications and applicability, has invested it with an importance and an interest which success seems only to stimulate and render more intense. The complexity of parts, and the diversity of combination, offer a scope for the exercise of ingenuity, alike highly inviting to the theoretical and the practical mechanic. The steam engine, even as a stationary power, is, moreover, of recent origin; and contemplating the phases which it has already assumed, in connection with the general feeling that its energies have not yet been fully developed, it is not matter of wonder that it has diverted attention from the less inviting problem which we are about to discuss. Water power is an old, if not an antiquated subject, on which the light of modern improvement has been but feebly reflected since the days of Smeaton. With a few exceptions, it has been abandoned to the management of those who recognise in it no principle, and no scope for improvement; and whose practice is not more opposed to improvement than it is empirical and opposite to all true principle. On the Continent, where water power is more valued than it is in this country, and where in general the means of feeding the energies of the steam engine are less abundant, the subject has indeed engaged the attention of some of the best minds of the age; but here, as in every department of the arts, the deductions of science are of little avail unless they be closely followed by an intelligent practice; and it cannot be denied that the learned and often meritorious investigations directed to this subject, which have issued from the French press, have been less successful in application than those directed to many other departments of constructive mechanics.

The fact that water power is an agency which cannot be augmented at pleasure, and which, in most situations where it is employed, has a full share of duty imposed upon it, renders it desirable that the best means of economising it be adopted. This implies a knowledge of some of the fundamental principles of hydraulics, in addition to that acquaintance with the general laws of mechanics which every engineer is assumed to possess. It is with the view of placing the subject in a distinct and concise form, and of pointing out precisely those principles which ought to guide the practice of the engineer in his dealings with this agency, that we undertake a brief exposition of the general problem. Under the title assumed, is implied the economy of water power, and the various means of rendering it available for purposes of industry. Without avoiding those abstract questions which beset the subject, and which imply some acquaintance with elementary analysis, we shall endeavour to keep in view that theory is valuable only in its relations to practice.

SECTION I.—CHARACTERISTIC VARIETIES OF WATER-WHEELS, AND THE THEORY OF THEIR ACTION.

By far the most numerous, and, therefore, important order of hydraulic movers, are those which come under the denomination of *vertical wheels*, from their movement being in vertical planes, and their axis of rotation consequently horizontal. Of these we have three varieties—named according to the points at which the water is received upon the periphery —*overshot*, *undershot*, and *breast wheel*. This last is further distinguished as *high* and *low breast*, according as the water is received upon the wheel above or below the horizontal plane of the axis. When the point of reception approaches the lowest point of the circle, it becomes an *undershot wheel*;

and when the water is *laid-on* near the summit and on the side which it approaches, the wheel is technically denominated a *pitch-back*. This differs from the *overshot wheel* in the direction of its motion, which is necessarily reverse to that of the stream; whereas, in the overshot wheel, the water, as the name implies, is carried over the summit of the wheel and discharged into the second or third bucket beyond the vertex on the opposite side.

To render these distinctions more apparent we may refer to the cut at p. 213 as an example of the common overshot wheel, strictly so called; and to the double Plate LXXXVIII.—LXXXIX. as a superior example of the modern pitch-back wheel. The example given in Plate XCIII. is that of a high-breast wheel. This differs, it will be observed, from the pitch-back wheel only in its being somewhat larger in proportion to the height of the fall to which it is applied. In Plate XCIV. we have an improved example of the undershot wheel, in which the mode of action of the water differs essentially from that in the other examples.

A second class of wheels, of which the varieties are even more numerous and hardly more distinct, comes under the denomination of *horizontal wheels*, because they move in planes parallel with the horizon, and have, therefore, their axes of rotation vertical. The primary types of this class differ nothing in principle from the undershot wheel, except that they are placed horizontally for convenience of applying their power. As the grinding of grain was one of the earliest purposes for which the motive power of water was employed, the horizontal arrangement was rendered necessary by the deficiency in the means of transferring motion and changing its direction by geering; and when the supply of water was large compared with the requirements of the district in which the mill was situated, it was not necessary to study economy in its use. The simplest mode of application was therefore adopted—which was to place the wheel horizontally; for in that position the millstones could be placed directly upon the vertical shaft of the wheel, thereby rendering any geering unnecessary. Every pair of stones had, of course, their own independent motor—which, in its most primitive form, consisted simply of a series of radial float-boards fixed upon the periphery of a strong wooden axle; and upon these the water acted by direct impulse as upon the floats of the common undershot wheel. When economy came to be considered, the impulse of the water began to be directed upon the floats by an ajutage contracting towards the aperture; and a further advance was to enclose the wheel in a cylindrical receptacle, to confine the action of the water upon the floats until it had yielded up some higher proportion of its moving force. This improvement, in its highest development, constitutes the modern *tub-wheel*, till very extensively used in some parts of the Continent and in America. It is, however, gradually disappearing before the numerous varieties of *turbine* which have of late been brought into competition with it. Of these the most reputed example is that of Fourneyron, who alone seems to have succeeded in realizing practically the results which theory indicates to be derivable by that species of wheel. In this country the more common varieties of horizontal wheels have not acquired any footing. The class is almost entirely represented by the *reaction wheel*—known in its primitive form, from the name of the inventor, as *Barker's Mill;* and as the capabilities of the principle of this wheel have been of late greatly developed under the patent of Messrs. Whitelaw and Stirrat, it seems to have at length obtained that attention to which, from its simplicity, it is entitled.

The varieties of this class of hydraulic movers which we have thought it desirable to exemplify are the two last:—the Reaction wheel of Whitelaw, Plates LXXXI.—LXXXIII.; and the Turbine of Fourneyron, Plates LXXXIV.—LXXXVII. These, in the present highly advanced state of the mechanical arts, when any degree of accuracy of workmanship can be obtained, appear to be the only varieties of practical value. They combine all the capabilities of the older forms, and admit of being constructed in better accordance with those principles of hydraulic action on which the efficiency of this class of wheels depends.

But before entering upon the question of the relative capabilities of these different machines, it will be necessary to indicate the measure of the agency from which their efficiency is in common derived. And, as a first step in this inquiry, it may be observed that, as there exists only one ultimate source from which the action of all hydraulic movers, of whatever description, is obtained, namely, the descent of a volume of water, by its own gravity, from a higher to a lower level, their action must have reference to the same measure, however differently it may be developed. Their common function consists simply in the interception of a *part*, and a *part only*, of the moving force of the fluid during its descent, and thereby rendering it capable of being transferred, by an appropriate system of organic connections, in an applicable form. Any estimate of the merits and capabilities of the different recipients must therefore have reference to the relation which the part intercepted bears to the whole force of the volume of water expended. It is this relation which is expressed by *percentage of effect*, and more concisely by *efficiency*—terms which are of constant use in the arithmetic of mechanical action.

In the action of any machine two systems of forces are concerned—the *moving* and the *resisting* forces. The machine is the medium through which these forces are maintained in equilibrium. Their magnitude may therefore be estimated statically; for the elements of resistance continually reproduced in one direction are balanced by the elements of pressure simultaneously reproduced in the opposite direction. The quantity of resisting pressure overcome in a given space is the *work done*. But as all pressure may be represented by weight and the overcoming of pressure by the elevation of an equivalent weight through a vertical space, the quantity of work will be measured and expressed by the product of the two numbers representing these quantities. The unit of work may therefore be taken = 1lb. raised to a height of 1 ft. in the unit of time. This definition is founded on the manifest assumption, that the weight remaining the same in every portion of the space, the power must be exerted afresh at every point through which the weight is raised. This does not directly apply when the weight is projected upwards by an impulsive force; for in that case the power ceases to act, and the body ascends to a height proportionate to the quantity of force accumulated in it at the instant of projection. But if a weight be raised by a rope over a pulley, so slowly that its inertia may be neglected, the force must continue to act constantly throughout the whole ascent; and if for an instant it be relaxed, the body begins to descend, in obedience to the force of gravity, unless prevented by some special contrivance. A new exertion is necessary to draw the body through every succeeding element of space: to raise it to a double height will therefore require twice the amount of power to be exerted; and to raise twice the weight will require the exertion of twice the power that would be required to raise the single weight to the same height. This proportionality holds for every other kind of work. If a certain amount of force be necessary to saw through an inch of a uniform block, an equal exertion will be requisite to saw through the next inch of the same block. In a saw-cut of a given depth and double length there are twice as many particles to separate, and therefore double the work performed as in a cut of single length. The cut of double length requires double the distance to be described by the force; and similarly in a cut of double depth, the number of particles separated in every element of space is double, and the resistance to be overcome is consequently twice as great. It therefore appears that the work is proportional to the resistance and the space conjoinedly, and may be measured by their product; so that if $\mathrm{P}=$ the quantity of resistance overcome expressed in lbs., and $s=$ the space through which it is overcome in feet per second, the effect realized is

$$\mathrm{E} = \mathrm{P}s \text{ units of work.}$$

The same measure applies when the resistance is variable; for if the force move through a given space while the resistances change through a series of n valves, we shall have as the *mean* pressure

$$\mathrm{P} = (\tfrac{1}{2}\mathrm{P}_0 + \mathrm{P}_1 + \mathrm{P}_2 + \mathrm{P}_3 + \text{-----} + \mathrm{P}_{n-1} + \tfrac{1}{2}\mathrm{P}_n)\tfrac{1}{n};$$

and, therefore, the work done over the space s will be

$$\mathrm{P}_s = (\tfrac{1}{2}\mathrm{P}_0 + \mathrm{P}_1 + \mathrm{P}_2 + \mathrm{P}_3 + \text{------} + \mathrm{P}_{n-1} + \tfrac{1}{2}\mathrm{P}_n)\tfrac{s}{n}.$$

The power expended is similarly measured by the weight and the space through which it has descended. If a weight $= w$ lbs. descend through a height $= \mathrm{H}$ ft. in a unit of time, the mechanical momentum is represented by the product $w\mathrm{H}$. But the height of a fall of water is independent of time, and therefore that the momentum may have reference to a fixed unit, the quantity w must express the weight of water which descends during that unit through the space H. A stream affording a double quantity of water, will therefore have as much momentum as another stream having a single quantity of water and double the height of fall—that is, $2w \times \mathrm{H} = w \times 2\mathrm{H}$. The height H is the interval of space from the higher to the lower levels, between which the moving force of the water is applied. This is termed the fall, and is either real or fictive: it is real when the fluid descends abruptly from the higher to the lower level; and is fictive when it flows in an inclined channel with a velocity $= \mathrm{V}$ ft. in a second. From the laws of dynamics we know that $\mathrm{V} = \sqrt{2g\mathrm{H}}$; and that the velocity, and the vertical height capable of generating that velocity, have therefore the relation $\mathrm{H} = \frac{\mathrm{V}^2}{2g} = 0{\cdot}0155\,\mathrm{V}^2$. These terms may be employed indifferently in the expression of the power; for if the value of either of them be known, that of the other may readily be found.

In this, as in all other estimates of mechanical power on a large scale, the gross unit of measure is the *horse-power*, of 550 lbs. lifted through a height of 1 foot in a second = 33,000 lbs. raised to the same height in a minute. To estimate, therefore, the quantity of moving force represented by a given volume of water descending from a known height, it is only necessary to divide the product $w\,\mathrm{H}$ by 550, or the product $\mathrm{W\,H}$ by 33,000, when $\mathrm{W} = 60\,w$. And if q be the number of cubic ft. of water supplied in a second, and Q the number supplied in the larger unit of time, we shall have $w = 62{\cdot}5\,q$, and $\mathrm{W} = 62{\cdot}5\,\mathrm{Q}$; and, therefore, the momentum of the fall will be expressed in units of horse-power by

$$\frac{62{\cdot}5\,q\,\mathrm{H}}{550} = \frac{q\,\mathrm{H}}{8{\cdot}8}\text{; and by }\frac{62{\cdot}5\,\mathrm{Q\,H}}{33{,}000} = \frac{\mathrm{QH}}{528}.$$

But in the application of power by means of mechanism, we in no case realize an effect equal to the expenditure. Confining our attention to the agency under consideration—the work done in no instance fully represents the momentum of the current. Some part of the water passes the machine inoperatively, and a part of the pressure due to the height of the fall is lost in consequence of imperfect adaptation. The velocity, and therefore the height due to the velocity, which the water possesses after it has passed through the machine, is likewise a deductive quantity. These circumstances, on which the degrees of economy of the different modes of application mainly depend, and with which we shall therefore have occasion subsequently to deal more particularly, prevent the transfer of a certain amount of the power of the water to the wheel. In the mechanism itself, whatever may be the arrangement of parts, there are besides absorbing influences which diminish the useful effect of the power actually transferred. Among these are the dead friction of the rubbing surfaces, and the tremour generally present from imperfect action, and which is propagated through the elastic materials of the supports, framing and foundations, to the ground by which it is absorbed. The loss sustained from this last is usually overlooked; but the power thereby lost is often of much more serious amount than would at first sight be supposed. When toothed geering is interposed to transmit the power to the working points, these influences are increased by the friction and shocks of the teeth—individually very small, but being constant and often repeated, the sum is always appreciable, and often of large amount.

When, therefore, the machine has attained to a permanent state of motion, the entire quantity of work performed during a given time is essentially less than the power expended by all the part which is non-effective; and the useful work is less than the effective power by the amount of all the passive resistances inherent in the mechanism. If, then, we put p to represent the weight equivalent to the useful or *active* resistances, and v the constant velocity with which they are overcome; also p' and v' to denote the same quantities in respect of the non-useful or *passive* resistances, we shall have as the whole dynamical effect

$$\mathrm{E} = p\,v + p'\,v'.$$

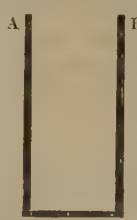
In words: the effect is equal to the sum of the resistances overcome. But as all the resistances are reducible to a common velocity v, we may put P to represent their entire sum; and if k be put to represent the non-effective part of the moving force, we shall have as the complete expressive

$$E = P\,v = w\,H - k.$$

In words: the effect is less than the momentum of the fall by the non-effective part $k = w\,H\,(1 - m)$. Whatever, therefore, may be the efficiency of the machine, we have the relation $E = m\,(w\,H)$, in which the coefficient m is a fraction, having its numerator equal to the effect, and its denominator equal to the power expended, that is, $m = \frac{E}{w\,H}$. This ratio can never $= 1$; but the machine will manifestly be the more perfect, the more nearly it approaches that limit; much less can m exceed 1, which would imply that the power realized is greater than that expended. The values which it bears in particular cases will be subsequently investigated, and therefore, in the meantime, it will be sufficient to observe that, in hydraulic movers of any kind, it perhaps never exceeds $\frac{4}{5}$; and in the undershot variety of wheels it is sometimes as low as $\frac{1}{4}$. The common range for wheels on falls of 7 ft. and upwards, is from $\frac{3}{5}$ to $\frac{3}{4}$.

The relation of the forces can therefore be generally defined, independently of the mode of action; but as the effects differ so widely, it may be presumed that they are in some degree influenced by the mode of application. Of this there are three forms: the fluid may act by its weight directly, by impulse, or by reaction. To these modes of development must be referred the *modus operandi* of all hydraulic movers; but it is very rarely that we find the effect dependent on a single mode of action. More generally two are concerned. In the first class of wheels, weight and impulse are almost invariably associated together, and not unfrequently in nearly equal degrees of intensity; and in the turbines of the second class, impulse and reaction are complicated to an extent which renders it often difficult to distinguish which influence most predominates. Centrifugal force is also an element which in most wheels requires to be appreciated; and in those varieties of the horizontal class which we have selected as examples, it enters as an element of the very highest importance. That these influences admit of a common measure of comparison has already been shown; they have a common origin, and are therefore distinguishable simply as modes of action of a particular force developed under special conditions. This can be rigorously established from general principles; but we shall arrive at our object more directly by regarding them as distinct elements—which they virtually become when reference is had to the methods of application.

Let attention be directed, in the first place, to the element of weight. If the weight of water $= w$ lb., descend from a height $= H$ feet in a second, it has already been shown that the mechanical momentum $= w\,H$ units. This is the maximum limit of the power which the stream can exert in overcoming resistance. But as an effect can no more fall short of, than it can exceed its producing cause, it may seem that whatever deficiency there is in the quantity of work done by this simple and direct mode of application, must be accidental. As the sum of all the momenta of all the resistances overcome, can never be less than the quantity of momentum expended by the water, it might seem practicable to realize, by some elementary contrivance, an effect equal to the expenditure. An equal effect, measured in work of the same kind, would be obtained by raising a volume of water equal to that expended, to the same height in the same time; and this, by sufficiently reducing the rate of working, is approximately possible, in the same manner and degree as it may be said to be possible that a weight suspended over a single pulley will, by its descent, cause the ascent of an equal weight to an equal height. But if the weights are absolutely equal, they will remain at rest unless externally influenced; and the more nearly they approach to equality, the smaller will be the quantity of motion produced in relation to time. The difference of weight must be constantly proportional to the quantity of work done, and this principle is general. Let there be two vertical cylinders which communicate with each other at their lower ends, and let water be supplied at A until it overflow at B; if then the supply be interrupted, there will be an equilibrium of pressures and the fluid will have the same surface-level in both cylinders. The columns being of equal heights will balance each other, irrespective of their volumes, in accordance with the well-understood principle that the vertical pressures are as the heights of the columns, and independent of their horizontal areas. This condition of equilibrium being established, if a stream of water be made to flow constantly upon the column at A, so slowly that the effect of inertia may be neglected, the fluid-level of the two columns will not be sensibly disturbed; and every particle of fluid which descends below the level at A, will cause an equal particle to ascend to the same level and overflow at B. If the action be regarded during the time that a lamina of the descending column passes from the surface to the bottom, and if μ be the weight of the fluid lamina, and H the height of the column, then will $\mu\,H =$ the momentum. But this is likewise the measure of the effect realized, for an equal lamina has meantime ascended to B through a height $= H$. And the same is true of all the other laminæ simultaneously in motion throughout any parts of the height H. The ratio of the work done to the power expended is therefore as 1 to 1, and it cannot possibly be greater; for every particle of fluid added at A can only displace a volume equal to its own at B; and it cannot raise an equal volume to a greater height, for then the surface level at B would exceed that at A, and the balance of pressures being thereby subverted, motion would ensue towards the lower level.



The maximum limit of the effect which can be produced is therefore represented by the measure of the momentum of the fall; and this limit is attainable only by virtually reducing the operation to the action, merely of the hydrostatic pressures. But when the rate of action is increased to a sensible velocity of the volume of fluid, the inertia can no longer be neglected. In illustration of this new condition, let it be assumed that every foot length of the descending column contains 100 lbs. of water, and that it is required to discharge a volume $= 400$ lbs. every second at B. This volume is $= 4$ feet in height of the acting column—which is therefore a measure of the velocity to be communicated to the fluid particles in the unit of time. Now it is a fundamental principle in dynamics, that the pressure necessary and sufficient to communicate a given increment of velocity $v = 4$ ft. per second to a given weight $w = 400$ lbs. of matter, is equal to the product formed by multiplying that weight into the ratio $\frac{v}{g}$ of the given accelaration v to that of gravity $g = 32{\cdot}2$. In the case supposed, the particles are uniformly accelarated from a state of rest to a velocity of 4 ft. a second: hence the pressure expended will be

$$\frac{v}{g}w = \frac{4}{32{\cdot}2} \times 400 \text{ lbs.} = 50 \text{ lbs. nearly.}$$

But when a body is uniformly accelerated from a state of rest, the space through which the pressure is exerted is equal to half the final velocity communicated: that is, the space $s = \frac{1}{2}\, v\, t$. The momentum expended in the unit of time $t = 1$ sec., in producing the required velocity will therefore be

$$\textit{pressure} \times s = \tfrac{1}{2}\, v \left(\frac{v}{g}\, w\right) = \frac{v^2}{2g} w = 100.$$

That is, a momentum of 100 must be constantly employed in producing the required velocity of 4 ft. a second in 400 lbs. of water starting from a state of rest. The surface levels of the columns must therefore stand at such heights that A—B $= h$; for the momentum due to the acceleration must be obtained by increasing the height of A or diminishing that of B. Now the quantity of water supplied in the unit of time at A $= 400$ lbs.; the momentum due to any height h of the column will therefore be $400\ h$; and as the whole momentum expended in producing the velocity is $100 = 400\ h$, therefore $h = \frac{1}{4}$ ft., the difference of height of the descending and ascending columns. This is irrelative of any frictional resistance in the apparatus to the motion of the fluid, and presumes that as soon as the particles ascend to B, they cease to press upon the succeeding particles or in any way to obstruct their motion. Both of these influences must be considered in practice and balanced by additional head pressure at A.

There is, therefore, a quantity of momentum $= 400 \times \frac{1}{4}$ ft. expended more apparently than is realized; but as every effect is necessarily equal to its producing cause, the deficiency must in some way be included in the result. This immediately appears from the conditions assumed; for, supposing the diameters of the columns to be equal, the water will be discharged with all the acquired velocity in it, and will therefore possess a projectile force sufficient, were all external influences excluded, to cause it to ascend to the height h', from which in falling freely it would acquire the given velocity of 4 ft. Now h' is known when the velocity v is known from the relation $h' = \frac{v}{2g}$; and the momentum $= wh'$ is therefore also known. By substituting the numbers of the particular case, we have $h' = \frac{1}{4}$ ft. and $wh' = 100$, which is identically equal to the momentum of the accelerating pressure, and $h = h'$.

It therefore appears that, considered dynamically, the whole effect produced is equal to the whole force expended; but if regard be had only to the quantity of water actually raised, and the height at which it is delivered, and which together constitute the measure of the useful work, a portion of the momentum of the fall equal to that expended in communicating motion, will not be realized. It is therefore obvious that if the fall of water be made to produce its effect by acting upon a machine which moves, at the point of application, with any constant velocity v ft. a second, a portion of the momentum of the fall, represented by $w\frac{v^2}{2g}$ will be retained by the water after it has acted, and the remaining part $w\left(\text{H}—\frac{v^2}{2g}\right)$ only will be expended effectively upon the machine.

This is the condition of those wheels on which the water descends vertically and acts directly by its weight. If the water enter the buckets with a velocity equal to the uniform velocity of the wheel, it will exert upon it a pressure equal to its whole weight; for as no acceleration takes place, no pressure will be expended in producing motion. The part of the fall which must be expended in communicating the initial velocity is represented by $h = \frac{v^2}{2g}$, and therefore varies with the velocity of the wheel; but as the velocity never exceeds 8 ft. a second, h will not exceed 1 ft. of the fall. The quantity of moving force impressed upon the wheel is otherwise independent of the velocity of the circumference, and is represented by the weight of water entering the buckets in the unit of time, and the space through which it descends before it is discharged from them.

To determine the effect when the water acts by impulse, let it be supposed that, instead of descending vertically, it is made to descend upon an inclined plane, and strike, with all the velocity which the fall can produce, upon the floatboards of an undershot wheel. The momentum of the fall will have the same measure as before; for all friction and other external influence being withdrawn, the water, on arriving at the lowest point of the descent, will have acquired an impetus sufficient to enable it to ascend an opposite incline, of proper form, to the height from which it descended. The height of fall producing the velocity, and the velocity produced will have the relation $h = \frac{v^2}{2g}$; and the momentum will be represented by $w\,\frac{v^2}{2g}$. This being the total momentum present, the question is reduced to finding the portion of it transferred to the wheel. To simplify the conditions as much as possible, let us suppose that the floats are radial and perpendicular to the direction of the current, and that they so completely occupy the water course that no important quantity of the fluid can pass without acting upon them. Under these conditions it is obvious that the motion of the floats must be less than the attainable velocity of the water; for, were they to move with an equal velocity, they could derive no impulse from it, and would be retarded by the fluid if they moved with a higher velocity. But if the floats recede before the impulse of the fluid with any given velocity v, the intensity of the impulse will be correspondingly lessened; for the recession is manifestly equivalent to an equal reduction of the velocity of the current. The effective velocity will therefore be reduced to $V - v$; and as the water must escape with a velocity not less than v, the momentum transferred to the wheel will therefore not exceed

$$\frac{w}{2g}\left\{V^2 - \{(V - v)^2 + v^2\}\right\} = \frac{w}{g}(V - v)\,v.$$

This reduced form of the expression may be established even more directly by observing that $\frac{w}{g}$ is the *mass* of the water which acts in the unit of time with the velocity $V - v$; the product $\frac{w}{g}(V - v)$ is therefore the pressure with which the wheel is impelled through the space of v ft. a second; but the mechanical effect, as before defined, is the product of the pressure into the space $= \frac{w}{g}(V - v)\,v$.

In this expression v is the only variable quantity. If $v = 0$, the expression vanishes and the effect is *zero*; for a machine which does not move transmits no power, whatever may be the pressure upon it. If $v = V$, the expression also vanishes, and the power transmitted is null; for when the wheel moves with the same velocity as the water, it can, as already observed, receive no impression from it, and can only move at that velocity when the load, including all the active and passive resistances, is nothing. Now, between the extremes $v = 0$ and $v = V$ there will be a value of v, for which $(V - v)\,v$ is a maximum. But if a number be divided into two parts, the product formed by multiplying together those parts will be the greatest possible when the factors are equal, and, therefore, in the present instance, when

$$V - v = v, \quad \text{whence } v = \tfrac{1}{2}V.$$

That is, the effect yielded by the wheel will be a maximum when the floats have a velocity equal to half the velocity of the current. If, therefore, we substitute $\frac{1}{2}V$ for v, in the general expression, we will have

$$\frac{w}{g}\left(V - \tfrac{1}{2}V\right)\tfrac{1}{2}V = \tfrac{1}{2}\,w\,\frac{V^2}{2g} = \tfrac{1}{2}\,w\,H,$$

when H is put for its equivalent $\frac{V^2}{2g}$. From this it then appears that the greatest possible effect derivable from a head of water acting by impulse is equal to half the momentum of the fall; and this maximum is attainable only on condition that there is no loss from frictional resistance in the channel, and that no particle of the fluid passes the wheel without producing its full effect upon the floats. If the wheel move in an indefinite volume of fluid, a portion of the effect will be absorbed by the lateral action of the water when its velocity is interrupted by the floats. Thus, if S be the mean area of immersion of the float, the quantity of water acting in the unit of time will be (62·5 S V) lbs.; and its pressure in the direction of the motion of the floats will be

$$n \, . \, \frac{62{\cdot}5\,S\,V}{g}\,(V - v).$$

In this expression n is a coefficient containing the corrections due to the lateral deviations of the filaments of the fluid when approaching the float, and which generally may be taken $= \frac{4}{5}$.

We shall hereafter find that these conclusions require modification; but it is sufficient in the meantime to intimate the mode of calculation, and to point out the theoretical conditions which form the basis of inquiry.

It now only remains in this place to indicate the general principle of the reaction of water. According to Newton's third law of motion, action and reaction are equal in amount and opposite in direction. This proposition assumes the character of an axiom, when the mind is directed to the reciprocal action of solids, since it is clear that any body acting upon another by pressure, for example, must itself experience a reaction equal and directly opposed to the action which it exercises. In the same manner, whenever a force, as that of gravity, pressing upon a fluid, causes it to issue through an orifice formed in the side of a containing vessel, a force equal and contrary to that with which the stream issues, will in consequence be expended upon the side of the vessel opposite to the orifice of escape. To explain this very briefly: when a part of the weight of a fluid is expended in producing motion in any direction, an equal pressure is necessarily deducted from its pressure in the opposite direction, for the gravitation employed in generating velocity cannot at the same instant be causing pressure. Supposing an orifice to be made in the bottom of a vessel filled with water, a column of the fluid will descend through it and will expend during its descent a quantity of pressure equal to a column of twice the depth of the fluid in the vessel, and having an area equal to the least section of the stream. For example: suppose the vessel to be 16 feet deep, and to be kept constantly full, the velocity of the stream will be 32 feet in a second; and, therefore, a column of 32 feet of length will pass through the orifice in each second, with the whole velocity derivable from its weight acting for the time. It is, therefore, clear that an equal amount of the pressure of the fluid in the vessel must be expended in producing that velocity, and must of course be deducted from the weight of the whole fluid—that is, from the entire pressure which it would otherwise exercise on the bottom of the vessel. Now, what is true with respect to vertical descent, is equally true when the motion is in any other direction. When the orifice is formed in the side of the vessel, the pressure upon that side will be diminished by as much as the pressure employed in producing the motion; and the effect of the diminution of the pressure in that direction will be the same as if the vessel were subjected to an equal pressure of any other kind in an opposite direction. And, moreover, the pressure being lateral, and therefore perpendicular to the only direction in which a vertical force, like that of gravity, can itself act, it must be derived from reaction of the opposite surface of the vessel upon the moving particles of the fluid, and may be assimilated to the constant pressure of a spring interposed between the moving particles and the part of the vessel immediately opposite to the orifice. In this position the spring must needs act in a direction exactly contrary to that of the movement impressed upon the fluid, and with an intensity exactly equal to the hydraulic pressure—that is, to the force due to the volume of water issuing by the orifice. Now, if s be taken to denote the cross sectional area (in square feet), of the stream, and h the depth of the water above the centre of the orifice, then the quantity of water discharged in a second will be $s\sqrt{2\,g\,h}$ cubic feet, and the weight $62.5\ s\sqrt{2\,g\,h}$.

But the hydraulic pressure due to this volume of water will be 62.5 $s \times 2\ h$, which is the weight in lbs. which would be necessary and sufficient to prevent the vessel from receding in a direction opposite to that in which the water issues. To approach the actual conditions: suppose a vertical cylinder with two hollow tubes inserted near its base, and projecting laterally at right angles to its axis; that these tubes are closed at their outward extremities, and communicate freely with the interior of the cylinder; that an orifice is pierced near the extremity of each on opposite sides of their common axis, and in a plane passing through that axis perpendicular to the axis of the cylinder; if this apparatus be placed on a vertical axis, round which it is free to revolve, it will constitute that variety of hydraulic machine known as Barker's Mill, and may be considered a type of those machines which derive their power from the reaction of fluids. Water being let in to fill the vertical cylinder, it will flow into the horizontal tubes and issue by the lateral orifices; but in thus finding vent into the atmosphere through the (contrary) sides of the tubes, these will be made to recede in a direction opposite to that in which the water flows out, and thereby produce a circular motion of the apparatus round the axis by which it is confined.

To arrive at a general notion of the power developed by the revolution of the machine, let us denote the depth of the cylinder above the level of the orifices by H, and the sum of the cross sectional areas of the jets by S; if the cylinder be kept constantly full of water to the depth H, then the weight which must be applied in an opposite direction to that in which the machine tends to revolve, and at the same distance from the axis of revolution as the centres of the orifices, to prevent the machine from getting into motion, will be $62\frac{1}{2} S \times 2H$ lbs., this being the hydraulic pressure due to the quantity of water $S \sqrt{2\,g\,H}$ cubic feet discharged each second. Otherwise expressed: the weight necessary and sufficient to balance the hydraulic pressure, and thereby to prevent the machine from revolving, is that of a column of water equal in length to twice the head, and having an area of base equal to the sum of the cross sectional areas of the two jets. This is found to agree with experiment, and it may be determined from *a priori* reasoning. In every body falling freely, the velocity acquired in a given unit of time is such as would carry it through double the space which it has fallen during the next equal unit of time, supposing gravity to cease to act upon it. There must, therefore, have issued by each of the orifices of the machine, a column of water equal to double the height of the surface above the orifices, that is 2 H, and the weight of such column is manifestly $62\frac{1}{2}\ S \times 2\ H$.

This will then be the condition of the machine held in a state of rest by a weight balancing the hydraulic pressure of the water discharged by its orifices. But when it is allowed to get into motion another important condition is superadded. Centrifugal force is brought into action, and increases the pressure of the water at the orifices, and thereby augments the quantity discharged in a given time, and also the intensity of the reaction, exactly as if the head pressure or depth of the cylinder were correspondingly increased.

A common expression for the centrifugal force of a body revolving in a circle at a distance x from the centre of motion is $\frac{w}{g}\ \omega^2 x$, in which ω is the angular velocity at the distance x, and is $= \frac{v}{x}$ when v expresses the absolute velocity in feet per second, at the distance x. Now if the mass $\frac{w}{g}$ of the body advance in the direction of the radius outwards, through the element of space $d\,x$, in an instant of time, the quantity of action *(vis viva)* created by the centrifugal force will be $\frac{w}{g}\ \omega^2 x\, d\, x$. But this being true of a solid body, will be equally true of the molecules of water in the arms of the machine. If, therefore, these arms commence at a distance r, and extend to a distance R (the centres of the orifices,) from the centre of rotation, we shall have, by integrating for the space between r and R, the length of each arm,

$$\int_r^R \frac{w}{g}\ \omega^2 x\, d\, x = \tfrac{1}{2} \frac{w}{g}\ \omega^2 (R^2 - r^2.)$$

And since $\omega = \frac{v}{R}$, if we take the quantity of water $w = 1$, we have, as the pressure at the orifices due to the velocity v of rotation.

$$\tfrac{1}{2} \frac{w}{g} \omega^2 (R^2 - r^2) = \frac{v^2}{2\,g}\left(1 - \frac{r^2}{R^2}\right).$$

If, therefore, we add this to the initial head pressure H, we shall have as the whole pressure at the orifices of the machine,

$$H + \frac{v^2}{2g}\left(1 - \frac{r^3}{R^2}\right)$$

Now, under this pressure, the expenditure of water will be increased as $\sqrt{2\,g\ H}$ to $\sqrt{2\,g\,H + v^2\left(1 - \frac{r^2}{R^2}\right)}$, that is, as 1 to $\sqrt{1 + \frac{v^2}{V^2}\left(1 - \frac{r}{R^2}\right)}$, putting V for the velocity due to the initial head H; and supposing the permanent head H to have been increased by the quantity $\frac{v^2}{2g}\left(1 - \frac{r^2}{R^2}\right)$, by directly increasing the depth of the cylinder, it is plain, from what has before been stated respecting the force of reaction, that the weight which would just be sufficient to keep the machine from getting into motion, would be

$$62.5\ S \times 2\left\{H + \frac{v^2}{2g}\left(1 - \frac{r^3}{R^2}\right)\right\} \text{lbs.}$$

This, then, is the whole pressure of reaction due to the increased head $H + \frac{v^2}{2g}\left(1 - \frac{r^3}{R^2}\right)$; but the reaction due to the fictive part $\frac{v^2}{2g}\left(1 - \frac{r^2}{R^2}\right)$ is obtained in consequence of the rotatory motion of the machine, with a velocity of v feet per second; a portion, therefore, of the whole reaction due to the quantity of water expended, must have been consumed in communicating that velocity to the volume of water discharged in that second of time. The pressure thereby withdrawn from that due to the condition of rest will be expressed by the mass × into the velocity, that is, by

$$\frac{62.5\,\mathrm{S}\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}}{g}\times v$$

And this pressure being subtracted from the pressure due to the whole force of reaction, there remains the whole effective pressure, that is, the resistance or load which the machine can overcome at the given velocity v feet per second. The operation being performed, we obtain

$$\frac{62\tfrac{1}{2}\,\mathrm{S}}{g}\left\{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)-v\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}\right\}$$

But the weight of water discharged in a second is

$$62\tfrac{1}{2}\,\mathrm{S}\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}=w;$$

therefore, $$62\tfrac{1}{2}\,\mathrm{S}=\frac{w}{\sqrt{\left\{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)\right\}}}$$

If, then, we put for $62\frac{1}{2}$ S in the foregoing expression this equivalent, and reduce, we obtain the convenient formula,

$$\frac{w}{g}\left\{\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}-v.\right\}$$

Now, this being the pressure acting upon the machine, and the velocity being v feet per second, the power transmitted supposing no loss, will be

$$\frac{w}{g}\left\{\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}-v\right\}v.$$

And putting $\mathrm{V}=\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}$, this formula takes the form

$$\frac{w}{g}(\mathrm{V}-v)v,$$

in which $w=$ the weight of water discharged in a second; V the velocity of the issuing jets, and v the absolute velocity of the machine, in feet per second. The theoretical rule thus agrees with that established for wheels which derive their efficiency from the impulse of the stream, thereby verifying the doctrine that action and reaction are equal. There still remains, however, to determine the value of the experimental co-efficient m, with which this expression must likewise be affected to render it available in practice; but this being different for differently constructed machines, we cannot pursue the inquiry in this place.

The rule may be otherwise established thus:—The whole labouring force, or mechanical efficiency of the water, expended under a head pressure of $\mathrm{H}+\frac{v^2}{2g}\left(1-\frac{r^2}{\mathrm{R}^2}\right)$ feet, will be

$$\frac{w}{2g}\left\{\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}\right\}^2=\frac{w}{2g}\left\{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)\right.$$

But of this efficiency there is consumed in giving rotatory motion to the water, and thereby raising the head pressure $\frac{v^2}{2g}\left(1-\frac{r^2}{\mathrm{R}^2}\right)$, the quantity $\frac{w}{2g}v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)$, which consequently falls to be deducted from the entire efficiency of the fluid.

Again, the water leaves the machine with a certain amount of velocity remaining in it, namely, a velocity equal to the difference between that with which it issues from the orifices under the virtual head $\mathrm{H}+\frac{v^2}{2g}\left(1-\frac{r^2}{\mathrm{R}^2}\right)$, and the velocity of the machine measured on the tangent to the circle through which the orifices revolve; this difference of velocity will be expressed by $\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}-v$, and the quantity of labouring force due to it will be

$$\frac{w}{2g}\left\{\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}-v\right\}^2$$

This likewise falls to be deducted from the labouring force due to the water expended under the head $\mathrm{H}+\frac{v^2}{2g}\left(1-\frac{r^2}{\mathrm{R}^2}\right)$ leaving the efficiency communicated to the machine

$$w\,\mathrm{H}-\frac{w}{2g}\left\{\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}-v\right\}^2=$$

$$\frac{w}{g}\left\{\sqrt{2g\,\mathrm{H}+v^2\left(1-\frac{r^2}{\mathrm{R}^2}\right)}1-v\right\}v,$$

which is the same formula as we obtained by the preceding investigation, and which, it may be well to observe, would correctly represent the action of the machine, were it not that it is liable to modification, by certain incidental influences, which remain to be examined when we come to treat of the details of construction, and other circumstances by which the efficiency of the machine is affected.

We now pass to the examination of the different varieties of wheels indicated at the beginning of this section, and shall take them nearly in the order given, but under a somewhat more convenient division.

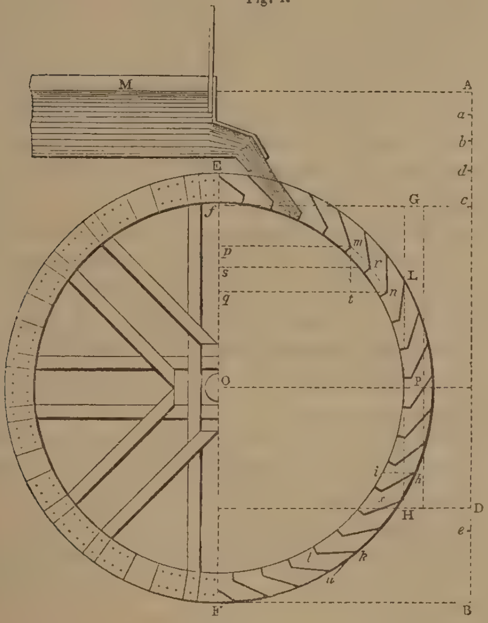
SECTION II.—BUCKET-WHEELS.

Under this head we include those nominal varieties of vertical wheels—overshot and breast—which are provided with buckets upon their peripheries for the reception of the water, and which, therefore, derive their efficiency chiefly from the weight of the fluid received into the buckets. This form of wheel, at whatever point the water may be received upon its circumference, is the most obvious in its action. No hydraulic machine could be more simple: a given weight descends from a given height: a known power is thus expended, to which the work performed ought to bear an assignable relation. The principle has long been universally recognized; and, therefore, the bucket-wheel, in its ruder forms, had an early place among contrivances for rendering water power subservient to the arts of industry; and, at the present day, it still maintains a precedence, to which, under certain limitations, it is justly entitled.

The older bucket wheels which we encounter, are constructed of wood; but that material, once of almost universal use in constructive mechanics, is fast giving place to iron, and in a few years hence we may expect that a wooden water-wheel will be as rare, and as much an object of antiquarian interest to those who take pleasure in reviewing the progress of the industrial arts, as wooden geering has already become. Many of those wheels still continue to exhibit in their constructive details, a very superior style of workmanship; and an attention to durability, which, in several instances within the knowledge of the writer, the lapse of a century has hardly conquered. The best specimens, no doubt, only remain of the truly old construction, while those of an inferior grade have disappeared and been replaced by wheels of modern construction, in which iron, if not the sole material, holds at least a prominent place.

Another peculiarity, not indeed uncommon in wheels of recent construction, although generally abandoned by millwrights who make pretensions to a superior knowledge of the principles which ought to govern the transmission of hydraulic power, unless the conditions be dictated by extraneous circumstances, consists in passing the water over the summit of the wheel into the buckets in the manner represented in the annexed figure. This arrangement constitutes properly an overshot wheel; but while we have preserved the name, it is no longer deemed necessary to apply it literally. In the present acceptation of the term, nothing more is implied than that the water is received into the buckets near the summit of the wheel; and, in ordinary practice, those wheels reckoned as overshot, by strict definition come under the designation of high-breast wheels. One of the finest specimens of this construction yet erected is that represented in Plate LXXXVIII.—LXXXIX., in which the height of the fall bears to the diameter of the wheel the relation of 9 to 10. In the purely overshot wheel this relation, as we shall immediately have occasion to show, is very nearly inverted.

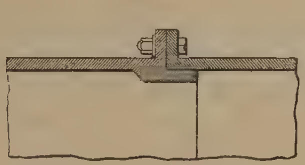
Fig. 1.



In the construction of wheels of this class, the technical points which fall to be considered, after determining the diameter and breadth of the wheel from the height of fall, and quantity of water furnished by the stream, are the axle and its journals, the arms and their connections, the shrouding, sole, and buckets.

The subject of the axle and journals has already been very fully noticed when treating of shafts in the article on Geering (pp. 132-3-4); but it may be here added that in iron wheels of great weight and breadth, in which the axle is consequently of corresponding diameter and length, and especially when the wheel is to be transported to a considerable distance from the work at which it is constructed, it is not uncommon to make the axle of two or even of three parts. When the axle is formed in this manner, the parts are fitted together by turning, and are secured by bolts in strong flanges cast upon the contiguous extremities. The annexed cut will convey an idea of this arrangement in its best and most enduring form—that in which it is least liable to objection. But no force of ingenuity can render it safe; the constantly changing position of the weight ultimately, and sometimes, indeed, very speedily, acts injuriously on the bolts, thereby relaxing the joint, which it is all but impossible to refit with any prospect of durability. This result is greatly delayed by boring the bolt-holes in the flanges, and turning the bolts exactly to fill them. In fitting the parts together permanently the bolts ought to be inserted hot and have the nuts fully screwed up before they have contracted to their normal length. As a further precaution, the bearing surfaces may be rusted together by washing them immediately before they are put together with a dilute solution of sal-ammoniac.

Fig. 2.

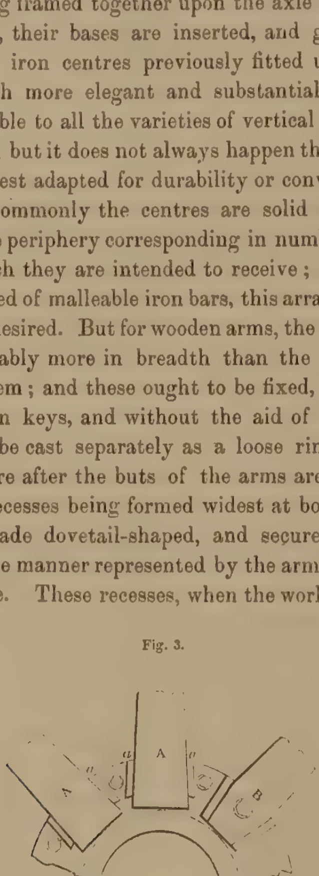


In wheels composed entirely of wood, the eight principal arms are commonly disposed in parallel pairs, crossing each upon the axle to which they have no positive attachment. The arms at the points of crossing are bolted together in sets of four, and the two frames thus formed are set apart upon the axle at a distance from each other, determined by the intended breadth of the wheel, and are bound together by tie-rods, and not unfrequently by diagonal struts when the breadth of the wheel is considerable, yet not sufficient to require the introduction of an intermediate set of arms and partition shrouding. The framing is further secured in its place upon the axle by wedging. The material used for this purpose is commonly oak, which, on being driven as firmly as the nature of the material will admit, is interspersed with thin iron wedges to give further compactness and solidity to the joint, and to prevent the packing from relaxing. These crosses of four arms each, thus fixed in position, sustain the two lateral shroudings between which the buckets of the wheel are inserted. They are termed the master, main, or principal arms; and in wheels of very small diameter—14 feet and under—there are no others employed. But when the diameter of the wheel exceeds the limit at which the arcs of the shrouding would be safely supported at that minimum number of points, a series of auxiliary arms are introduced, in sets of four, on each side of the wheel. These secondary arms do not cross each other at the axle as the principal arms do, but are simply made to abut against its faces, where they are secured by filling blocks, laid in in different ways, according to the number of auxiliaries introduced. They are further secured by bolts to the master arms, which they are always made to cross in the manner represented in fig. 1.

When the wood is sound, and no particular circumstances intervene to increase the strain upon the wheel, the strength of the arms may be computed by the ordinary rules applicable to the kind of wood employed in the construction. In a very elegant specimen of 48 feet diameter the arms at the base are 8 inches square, and taper to 6 inches at the extremities.

In wooden wheels of more modern construction, instead of the arms being framed together upon the axle in the manner just described, their bases are inserted, and generally fixed by wedges in iron centres previously fitted upon the axle. This is a much more elegant and substantial arrangement, and is applicable to all the varieties of vertical wheels, and to all diameters; but it does not always happen that the mode of fitting is the best adapted for durability or convenience of repair. Very commonly the centres are solid castings, with recesses in the periphery corresponding in number and size to the arms which they are intended to receive; and when the arms are formed of malleable iron bars, this arrangement is all that could be desired. But for wooden arms, the recesses ought to be considerably more in breadth than the buts to be inserted into them; and these ought to be fixed, not with iron, but by wooden keys, and without the aid of bolts. If the outside cover be cast separately as a loose ring, and bolted upon the centre after the buts of the arms are fitted, it will allow of the recesses being formed widest at bottom, and the buts being made dovetail-shaped, and secured by parallel keys *a a*, in the manner represented by the arms A, A, in the annexed figure. These recesses, when the work is of a superior character, would be planed on all the bearing surfaces, and the cover might be fitted by turning.

Fig. 3.



This mode of fitting is equally suitable for cast-iron arms, except that the keys are in that case rendered unnecessary by the buts being made to fill the recesses, and are carefully fitted by planing. They are also secured by bolts, and it is not often that any cover is employed.

When the wheel is of small diameter, the centres and arms are sometimes, and advantageously, formed of one casting. In this case no fitting is required; and although the moulder's labour is greater, there is an ultimate economy, when the diameter does not much exceed 12 feet, which more than balances the excess of foundry cost. In one small example, we have observed the principle extended to the shrouding; each side of the wheel consisted of a single casting, and the two were simply bound together by a few tie-rods, and the sheet-iron plates, of which the buckets were composed.

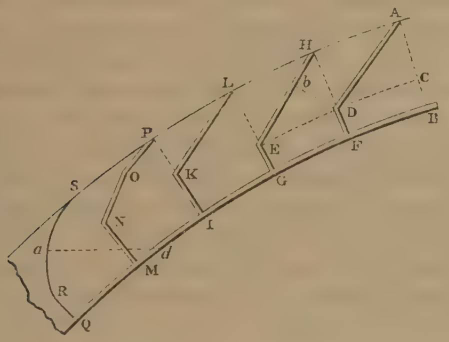
The shrouding is formed of two annular plates, which, in wooden wheels, are composed of plank, of 3½ inches to 7 inches thick, shaped and jointed together similarly to the felloes of a carriage-wheel. Instead, however, of the joints being formed by simply abutting the extremity of one piece upon that adjacent, the extremities are checked to half their thickness upon opposite sides, and overlapped. Sometimes also the joints are made by scarfing, when not opposite to the arms, and strengthened by plates of iron laid into recesses flush with the general surface. The other joints which receive the extremities of the arms, are half-checked on the exterior surface, and are connected by the palms, which are usually counter-checked on the inner face, to fall flush into the recesses prepared for them in the ends of the pieces which they are intended to connect. With the old millwrights it was not uncommon to form the joints of the better class of wheels by mortice-and-tenon, and sometimes by joggles—a method which seems to us equally efficient and less expensive.

In the older wooden wheels, the shrouding is usually of a greater depth than is consistent with modern practice. At present we meet with few examples in which the shrouding is more than 12 to 15 inches deep, whereas a depth of 20 inches was formerly not uncommon. This is a subject, however, which will require to be considered subsequently, as also the width of the wheel or distance between the shrouds.

The width of the wheel determines the length of the buckets. These extend between the shrouds, except in very broad wheels, in which an intermediate or partition shrouding is sometimes introduced to give support to the buckets at the middle of their length. This is usually resorted to in wooden wheels when the breadth exceeds 8 feet, but it is not uncommon to find iron wheels of double that breadth without any partition shrouding. In this class of wheels it is, indeed, an excepted case in which we find this arrangement adopted in the entirety exhibited by the older specimens. In these, the partition shrouding was, in all respects, a duplicate of the lateral shrouds, and was like them attached upon arms radiating from an independent centre, placed intermediate to the two others upon the axle of the wheel. Besides giving additional rigidity to the structure, it served to carry the medial extremities of the buckets, which being formed of plank, did not admit of that ready mode of staying, so convenient with sheet-iron buckets, as exemplified in Plates LXXXVIII. to XCIII. The second of these examples, is 22 feet in breadth, yet there is no partition shrouding, but the connections are stiffened, and the buckets supported at the middle of their length, by a cast iron ring, to which the diagonal arms are attached, as explained in the descriptions of the Plates, page 80.

The shrouds being properly adjusted and fixed in relation to the axis of motion, a circular runner of plank having a transverse section of 3 × 3 inches, is attached by wood-screws to the interior face of each of the shroud-plates, and upon this the ends of the sole-planks are supported. We speak of the best form of construction; but more frequently, the sole-planks are simply fitted and fixed upon the interior circumference of the shrouding; and this arrangement has its advantage in the facility which it allows for the repair of the sole, as the planks may be taken off without disturbing the buckets in their vicinity. When the sole is completed the wheel has the appearance of a large drum with radial flanges, between which the buckets are to be fixed. The extremities of the planks of which the buckets are formed, are commonly received into mortices cut in the contiguous faces of the shroud-plates; or into grooves formed by narrow runners of wood nailed upon the plates. A still better mode of fitting is to sprig the buckets in their places, or to mark off their form and positions upon the inner surfaces of the shrouding, and fill in the spaces between them by plates of board cut to the proper form. When the first method is adopted it is necessary to insert the radial part DF of the annexed fig. 4, before fixing the sole planks; but by forming grooves according to the latter methods, the sole may be completed previous to any part of the bucketting being prepared. When the method of mortising is adopted it is, on the contrary, more convenient to make the bucketting precede the application of the sole-planking. It is of little moment which of these methods be adopted in practice; and the circumstances of the particular case will invariably determine that which may be resorted to with most advantage; but it is of the utmost importance that in the process of *cleeding* the wheel be not thrown out of truth. To avoid this, the operations ought to proceed from at least two points; and if the wheel be of large diameter, it will give a better chance of correctness to distribute the operations over the four quadrants of the wheel, taking alternately that immediately opposite and contiguous as the next in succession to be operated upon.

Fig. 4.



Each bucket consists usually of two plates placed at a determinate angle ADF, fig. 4; and sometimes of three, as in the form indicated at PM. The bucket is an essential part of the wheel, which ought to be determined, not by the rule-of-thumb practice admissible in some of the less important details, but from a competent regard to the conditions involved. Upon their form depends, in a great measure, the efficiency of the wheel as a hydraulic motor; and although with wood the true conditions which lead to the best effect can only be very distantly approximated, it may still be of advantage to indicate these as nearly as the circumstances will admit.

Having determined the diameter of the wheel and the depth of the shrouding—the rules for which will be hereafter adverted to at length—let portions of their circles AS and BQ, fig. 4, be described from a common centre, as O in fig. 1. Let the depth AB of the buckets be divided into three equal parts and with a radius to C, making BC = $\frac{1}{2}$ AC, describe the arc CDE of the third circumference upon which the centre of gravity of the water contained in the buckets will always be found, at least very nearly. This radius, marked *Or* in fig. 1. is termed the *dynamic radius* of the wheel, and it is of importance that it be correctly defined. The distance of one bucket from that succeeding it, is measured upon this last circumference, and may be taken generally at 12 to 16½ inches. The common practice is to divide the circumference into four equal parts, and in each quarter is inserted the same number of buckets, the distance being made to vary slightly with the size of the wheel. But it also follows from this practice that the number of buckets will not be exactly proportional to the diameter. An approximate rule is to double the number expressing the diameter of the wheel in feet, and call that or the next higher number divisible by 4, the number of buckets.

Thus the diameter being 12 ft., the number of buckets	24
17	36
21	44
25	52

In greater diameters, the proportion increases thus,

diameter, 28 ft.,—	buckets, 60	diameter, 40 ft.,—	buckets, 92
32	72	42	96
38	84	46	108

We do not instance these as examples to be copied; but simply in illustration of a practice not yet quite obliterated, of referring the most vital conditions of the question to a hap-hazard empiricism which pretends to no better foundation than its affording an easy approximation. The proper elements from which the capacity of the buckets ought to be determined are the quantity of water to be used, and the angular velocity of the wheel, subjects which will be subsequently considered in relation to this and other questions of equal importance.

The circumference described by the dynamical radius of the wheel being divided into equal parts, C, D, E, &c., in number equal to the number of buckets which the wheel is intended to have, the radial lines CB, DF, EG, &c., will determine the direction and breadth of the *starts* of the buckets. The *floats* AD, HE, &c., are determined by regard to the condition that the angle HEG comprised between the start and float of the same bucket, ought to open as little as possible in order that the bucket may retain its water as long as possible, but at the same time the angle must be sufficiently great that the water shall have free admission through the space D*b*, otherwise a portion of it would be thrown back chiefly by the action of the air confined in the bucket—an evil now almost entirely remedied by the system of ventilation introduced in the construction, but which in wooden wheels could only be effected very imperfectly. In all cases it is essentially and obviously requisite that the space between the floats be considerably greater than the thickness of the sheet of water which is thrown upon the wheel. It is true that by increasing the breadth of the stream, its thickness may be diminished at pleasure; still it is necessary to give to D*b* a minimum of 5 to 6 inches to insure free ingress to the water in all positions of the bucket while receiving its charge. This condition is attained by giving to the angle HEG a value of 110° to 118°, supposing the wheel to have a diameter not less than 15 feet. Under this arrangement the float will have an inclination to the interior circumference—rather to a tangent drawn to that circumference from the point of intersection with the exterior circumference—of about 31° and not more than 33°. This angle is indicated by the tangent LH in fig. 1.

Some millwrights have endeavoured to lessen the evil experienced in the water being thrown out of the buckets by increasing the breadth of the starts, as in the bucket LKI, in which IK = ½ IP. And as an approach to the more modern curved bucket, the form PM has been employed, in which the float is composed of two parts, PO, ON, joined together at an angle of about 150°. This form has an advantage in so far as it retains the water longer; but besides increasing the labour and difficulty of fitting and repairing the buckets, it has the further effect of contracting the space through which the water is received into the bucket, which under the very imperfect system of ventilation practicable in wood wheels, is an evil more than equivalent to the advantage secured. Even the mode of fitting the two plates of the bucket together by a bevil-joint, as at D, is considered by some millwrights as an unnecessary technical difficulty which they avoid by fitting the start-board at an angle of 90° with the float board. This practice has an advantage in point of simplicity of construction, and it does not seem liable to any serious theoretical objection. But the difficulty complained of is more effectually removed by the introduction of sheet-iron buckets, which is not now uncommon, even in wheels principally constructed of wood. The usual practice is to curve the plate into the form SQ, in which the start to the point R on the dynamical circumference approaches the circumference of the wheel at an angle of 90°, and the extremity S meets that circumference at an angle of not more than 3°. In this mode of bucketting, which when introduced was reckoned an improvement of much value, and which in modern practice is the simplest that can be adopted, the buckets carry their water to the lowest point of the revolution of the wheel; but like that previously noticed, although it has an advantage in this respect, it has been considered liable to the objection of affording a less ready admission to the water in filling. In early practice, and indeed until very recently, the necessity of facilitating the escape of the air from the buckets when receiving their charge of water at the summit, was not understood. The importance of making the buckets of a form to carry the water to the lowest point of the revolution of the wheel was obvious; but when this was accomplished, it was usually found that the wheel neither received nor parted with the water freely. The late Professor Robinson seems to have been the first to give a full explanation of these circumstances, and to have pointed out at least a partial remedy. In reference to the first, he observes that "the half-taught millwright attempts to retain the water a longer time in the buckets, but finds that it gets into them with a difficulty for which he cannot account, and spills it about even after they have ceased to receive any from the spout. This arises from the air which must find its way out to admit the water, but is obstructed by the entering water, and occasions a great spluttering at the entry. This obstruction is vastly greater than one would imagine, for the water drags along with it a great quantity of air, as is evident in the water-blast described by many authors."

After observing that "this evil may be entirely prevented by making the spout considerably narrower than the wheel, and thereby leaving room at the two ends of the buckets for the escape of the air," he proceeds to consider the circumstances attending the emptying of the buckets. He observes —"there is another very serious obstruction to the motion of an overshot or bucketted wheel, especially when it revolves in backwater. It is not much resisted by the water on account of the slowness of its motion; but it lifts a great deal of the water in the rising buckets. In some particular states of backwater the descending bucket fills itself completely with water; and, in other cases, it contains a very considerable quantity of air of common density, while in some rare cases it contains both water and air in a compressed state. In the first case, the rising bucket must come up filled with water, which it cannot drop till its mouth gets out of the (tail) water. In the second case, part of the water goes out before this; but the air rarifies, and therefore there is still some water dragged or lifted up by the wheel, but (which is as detrimental to its performance) the descending side is employed in condensing the air; and although this air aids the ascent of the rising side, it does not aid it so much as it impedes the descending side, being (by the form of the bucket) nearer to the vertical line drawn through the axis."

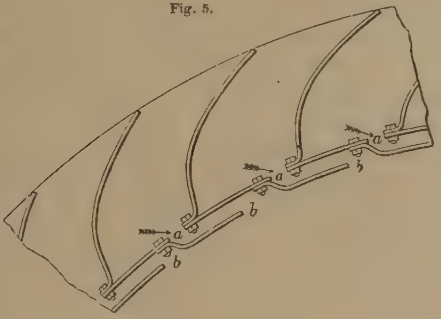
Without acquiescing in the correctness of the objection and explanation contained in this second case, it is not difficult to perceive that if the bucket be quite air-tight, and of such a form as to carry its charge of water to the bottom of the circle of revolution, in the process of emptying itself against the atmospheric pressure, and in a direction contrary to the direction of motion of the wheel, there will be a partial vacuum formed in the bucket, and to that extent the effect of the wheel will be diminished. If the bottom segment of the wheel be immersed in tail water, which is a very frequent circumstance, the evil will be greatly increased; for then the bucket cannot relieve itself until its mouth has ascended above the water-level; and in consequence of the resistance offered to the descent of the water, the bucket will have ascended considerably higher before it is entirely relieved of its load.

The earliest remedy applied to the removal of these difficulties, and which has been continued to some extent to the present time, was to bore holes in the starts of the buckets. As respects the admission of the water, this contrivance is of little value, as the escape of the contained air by the holes is prevented by the presence of the water in the inferior bucket. But its success in promoting the easy discharge of the water from the buckets soon became obvious from the increased efficiency of the wheels in which it was adopted.

And besides its effect in facilitating the relief of the buckets by the admission of the air to replace the water discharged, it has the further advantage of preventing the troublesome and often dangerous effect of *bucketting* when the works are stopped, as the small quantity of water escaping by the sluice and falling upon the wheel runs through the air-holes without accumulating in the buckets of the wheel, and starting it into motion at intervals.

Mr. Fairbairn of Manchester, to whom we owe the present improved system of ventilation, relates, in a paper submitted to the Institution of Civil Engineers, that about twenty years ago he constructed a wheel for Mr. James Brown of Linwood, near Paisley, "which in floodwaters, when the wheel was loaded, every bucket acted as a water-blast, and threw the water and spray to a height of six to eight feet above the orifice at which it entered. This was complained of as a great evil; and in order to get rid of the difficulty, incisions were made through the sole plates, and small interior buckets were attached to the inner sole plates, as represented in the annexed sketch. The air made its escape by the openings *a a a* into the interior buckets *b b b* inside of the wheel during the time of filling; and, when working in backwater, it presented the same facilities for its emission before even a partial vacuum could be formed in the ascending bucket when rising through the tailwater. The changes which this remarkable alteration effected can scarcely be credited; the wheel not only took and parted with the water with perfect freedom, but an increase of power of nearly a fourth was obtained. The wheel is still in the same state, and continues in all states of the water to perform an apparent and satisfactory amount of duty."

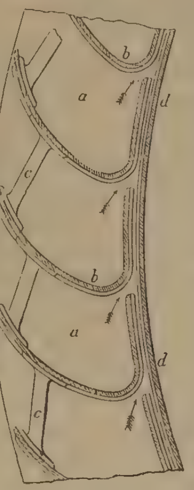
Fig. 5.



This was an important advance on the scheme of piercing the starts of the buckets; and although in this case applied to an iron wheel, it admits of easy application to wooden wheels, by making a species of internal sole and dividing it off into portions answering to the small buckets *b b b*, and of course equal in number to the number of main buckets in the wheel.

The improvement effected upon Mr. Brown's wheel subsequently induced Mr. Fairbairn to extend the principle of ventilation; and instead of waiting to ascertain the action of the wheel when started, to apply it as a fundamental requisite to be provided for in the construction. The primary form of the contrivance, although very effective, is manifestly liable to objection in respect of the additional workmanship which it required; and besides, if thoroughly carried into effect, would tend to weaken the sole plate of the wheel. To obviate these objections, Mr. Fairbairn introduced the elegant arrangement depicted in the annexed cut, by which the integrity of the sole is preserved, and in which, with very little additional cost of construction, the object is most effectually accomplished. The method consists in forming the buckets—each consisting of a single plate—with independent soles, which are applied parallel with the sole of the wheel, leaving between the two contiguous surfaces a vacant space of about an inch, for the escape of the air into the superior bucket during the process of filling, and which will obviously serve to readmit the air to replace the water when the bucket begins to empty itself towards the lowest point of its revolution and begins to ascend. When it has attained this position, it ought manifestly to be entirely relieved of its burden; but it has been already intimated, that, in certain conditions of backwater, this can be accomplished only by some contrivance for admitting the pressure of the atmosphere upon the water in the bucket immediately on its beginning to ascend. If the atmosphere be then excluded, and the bucket be full of water, a certain amount of weight will be made to act adversely to the motion of the wheel, and to that extent will diminish its efficiency. This takes place more commonly with iron than with wooden buckets destitute of provision for their ventilation, on account of the more contracted form of the inlet passages produced by the curvature of the floats of the former.

Fig. 6.



On this subject Mr. Fairbairn remarks, that "water wheels constructed entirely of iron, and having thin plates instead of wood for the buckets, give decreased facilities for the admission of the water, and for the escape of the air contained in them. So great difficulty is experienced in effecting the discharge of the air in a close bucket through the same orifice, and at the same time that the water is being admitted, as in some wheels almost entirely to prevent the entrance of the water; and in cases where the buckets are closely formed, the wheel is deprived of almost half its power by the reaction of the inclosed air. This defect is most obvious in water wheels having contracted openings—which may be easily accounted for in every case where the water is discharged upon the wheel in a larger section than the opening between the buckets. Under these circumstances the air is suddenly condensed, and again reacting by its elastic force, throws back the water upon the orifice of the cistern and thus allows the buckets to pass without being more than half filled."

These remarks are intended to show the extent of the difficulties which have been removed by the introduction of the ventilated bucket, and to induce the adoption of that system, which, we may remark, is always applicable, and presents none of those constructive objections which are so often fatal to the introduction of technical improvements in hydraulic machinery. This has indeed been acknowledged by the almost universal adoption of the principle by those engineers capable of appreciating its advantages; but examples in which it has been disregarded—possibly through ignorance of its use—have fallen under our notice very recently, and in more than one instance very fully illustrated the danger of dealing with hydraulic power on mere empirical rules.

When the wheel is wholly constructed of iron, the buckets are usually supported at their extremities on narrow flanges cast of the intended form upon the inside faces of the shrouding, and secured by small bolts (usually half inch in diameter), for which the holes are cast in the flanges, and pierced in the bucket plates. They are further supported upon each other by intermediate stays, cast with palms of opposite curvature at their extremities, to meet the interior and exterior surfaces of the buckets which they are intended to connect. The details are fully illustrated by the drawings of Plate XCI., of which a very minute description is given at pp. 79—85. It is, however, to be remarked, that, except in the undershot wheel of M. Poncelet, no attempt has hitherto been made to give the buckets a definite form with reference to the action of the water upon them on its admission ; and possibly under the system now generally adopted, of introducing it below the summit and under as small a head pressure as can be obtained, consistently with the volume to be used, it is of little importance to bring that element into the calculation. It may, nevertheless, be observed that, in strictness, when the water is allowed to fall simply over the lip of the bucket, the curve ought to be that of quickest descent; and in no case ought it to descend from so great a height that its reaction upon the underside of the succeeding bucket shall be sensibly felt. This appears to be well illustrated by the well-known experiments of Mr. Smeaton, to which we shall hereafter have occasion more particularly to advert. His overshot model was 24 inches in diameter ; and when the whole descent of the water was 27 inches, the maximum effect of the wheel was 76 per cent. of the power of the water; but when increased to 35 inches, the ratio fell to 52. In other words, the head being increased in the ratio of 7 to 9, the increase of effect is only in the ratio of 27 to 28, and, consequently, the increase of effect is not $\frac{1}{7}$th of the increase of perpendicular height. From this he concludes, and correctly, as respects purely overshot wheels, " that the higher the wheel is in proportion to the whole descent, the greater will be the whole effect." But the explanation which he offers is founded entirely on the fact—" that the effect of the same quantity of water, descending through the same perpendicular height, is *double* when acting by its gravity upon an overshot wheel, to what the same produces when acting by its impulse." It is unnecessary, in the meantime, to examine this proposition, as it is sufficient for our present purpose to intimate that a high velocity of the current entering the buckets is attended with a loss of effect, and that at least, a portion of that loss seems to us to result from the reaction of the water against the sole of the bucket into which it is received, and against the bottom of the next succeeding bucket ; and it is obvious that any impulsive force expended on these surfaces must proportionally resist the motion of the wheel in the contrary direction.

As respects the mode of supplying the water to the overshot wheel, the arrangement is sometimes made to differ slightly, according as the level of the water in the reservoir or dam is nearly constant, or varies between wide limits. In the first case, especially if the dam be close at hand and the duty of the wheel nearly constant, the channel is simply formed of the proper width to bring forward the maximum quantity which is intended to be employed upon the wheel; and a sluice is placed at the origin of the channel in the dam breast to regulate the quantity drawn off for immediate use. From this sluice, the channel—often formed of wooden troues—follows the most direct line over the summit of the wheel, and is terminated by a spout inclined at an angle sufficient to throw the water perpendicularly upon the start of the second or third bucket counted from the summit. The *apron* or sole of this spout is usually from 18 to 24 inches long, and declines from the bottom level of the channel at an angle of 12 to 18 degrees, forming an inclined plane on which the water attains the required velocity before entering the buckets, and which ought to be at least equal to that of the circumference of the wheel, otherwise, it is struck, and some portion of it thrown off by the float-boards of the buckets as they successively come into position with the current.

When the dam is at some considerable distance—rendering it inconvenient to have recourse to the sluice there situated on every occasion that it may be necessary to modify the power of the wheel, a second sluice or *shuttle* is placed contiguous to the spout, and which, in the common class of wheels, is usually set by hand at the required height, often directly, but sometimes a simple contrivance, consisting of a weighted lever intervenes, by which the shuttle can be adjusted by a cord brought inside of the building. In the higher class of wheels, and especially when great steadiness of motion is required, the shuttle is worked by a self-acting apparatus—shown in its most complete form, with all the improvements and appliances of modern mechanism, as applied to the large wheel at Greenock, in Plate XC.

When the course is of considerable length, and the level of the water in the reservoir or dam is subject to sudden variations, it is of advantage to adopt a slight modification of the spout by which the water is thrown upon the wheel. If the sluice in the dam breast be set to furnish the proper supply of water under a given head, and the level be increased, it will, of course, discharge a greater quantity than is required, and thereby produce an increased head-pressure upon the shuttle. This may be lowered to diminish the quantity thrown upon the wheel ; but, in consequence of the head-pressure accumulated in the course, the velocity of efflux would be increased often to such an extent as, without precaution, to throw the water entirely over the buckets. To prevent this, and, at the same time, to take advantage of the increased impulse of the water, the spout is provided with a cover inclined to the axis of the orifice, and connected, water-tight, with the back plate of the shuttle. The spout has thus the outline of a truncated pyramid, of which the faces converge towards the extremity at an angle of 6 to 7 degrees with the axis, which is directed towards the superior surface of the start of the bucket immediately in advance of it. The direction of the current upon the wheel is thus preserved under any variations of head pressure upon the shuttle; and the impulse being directed as nearly as possible in the line of motion of the point impelled, its value becomes an increment to the force of gravity of the water, and, to some extent, economizes a power which would otherwise act injuriously in projecting the water beyond the proper range. The horizontal dimension of the orifice, under this arrangement, and especially when the system of ventilation is incomplete, ought to be a little less than that of the buckets and the height perpendicular to the axis ought not, in general, to be more than four inches, and it is almost always better to be less.

This form of spout is represented in Fig. 1. It has not been in very general use in this country, although examples are still met with; but it is still very commonly employed on the continent, especially in France, where it bears the significant name of "duck's bill." In some cases, also, the spout is attached to the feed-box or small reservoir formed over the wheel in some works, where the water is brought forward from the main dam by a large pipe or covered conduit passing along the surface of some intervening ground situated on a lower level than the summit of the wheel, and sometimes entirely under ground, when it is necessary to keep the surface free of interruption for any particular reason, as the crossing of a road and the like. In this arrangement there is always a certain amount of loss of head, incurred by the passage of the water through the conduit, which must be taken into account in the construction of the wheel. The effect of this diminution of head pressure is to prevent the water from rising to the same level in the feed-box as in the reservoir, by diminishing the force of the current, and this loss—which is also a loss of fall upon the wheel, and, consequently, a loss of moving force, without any compensating advantage, except the convenience afforded under particular circumstances—must be deducted from the height of the fall in determining the diameter of the wheel.

The natural situation for the opening to the culvert for carrying away the tail-water is on the side opposite to that at which the water is received by the wheel; for, in that case, should the lower arc of the wheel dip, which it almost invariably does, from the desire entertained by the millwright to economize the fall as much as possible, the run of the water will impede it less than if its motion were opposed to the motion of the wheel. When the wheel is kept entirely above the level of the tail-water, this arrangement may, of course, be reversed; it is then immaterial at what point the culvert opens into the wheel-pit, except that the water will rise higher when the direction of its motion is changed from that impressed upon it by the motion of the wheel, and, consequently, a greater amount of tail-clearance must be allowed when the escape is retarded by a change of movement. This is, therefore, always avoided as much as possible; but, in situations where the lead of the water is determined by circumstances of locality, and the direction of the motion of the wheel by circumstances of convenience, we commonly find that when these conditions conspire to render the common arrangement inapplicable consistently with economy, the water, instead of being led over the summit of the wheel, is thrown upon that side of it to which the current approaches, the spout of the shuttle being deflected sufficiently backwards to reverse the motion of the current, and direct it upon the circumference at some distance below the summit. It is easy to perceive that this simple arrangement, which at first was resorted to rather as an artifice to avoid an inconvenience, very severely felt in cases where the direction of the waste water culvert was fixed by local circumstances, and in which the wheel-pit was liable to be flooded with backwater, allows of the wheel being made of any desired diameter even greater than the height of the fall. It also requires less of the head to be sacrificed between the shuttle and the point of the wheel at which the water is received, and is, therefore, more economical than the primary arrangement from which it was considered a deviation merely allowable in obedience to the local conditions of the particular case. Its advantages, however, soon began to be perceived, especially in low falls, and examples of its application speedily became numerous. This was favoured by the theoretical notion, that a wheel of large diameter is, in all cases, the most economical; and, as already observed, this arrangement gives free scope to the millwright to adopt any diameter of wheel he may think fit. The system has now become general, and constitutes one of the main characteristic features of the modern bucketted wheel. It has, besides, removed the distinction between overshot and breast wheels which was formerly significant. The term overshot is, indeed, not now applicable to any of our larger modern examples, except by a forced interpretation of the term; and, instead, we ought, in strictness, to employ the term high-breast as better expressing the actual conditions. We have still, indeed, some wheels of a minor class scattered over the country and employed chiefly for agricultural purposes, which belong to the primitive order; but they are only retained in situations where the power is superabundant, and where it is, therefore, not necessary to look narrowly into the economy of its employment.

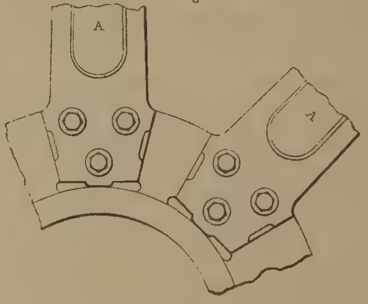
In the higher class of wheels constructed according to this principle—and which were originally denominated *breech-wheels*—the shuttle consists of an accurately fitted sluice, usually commanded by an adjusting apparatus, bearing the name of *governor*, which regulates the supply of water admitted upon the wheel to the power required. The technical details and mode of action of this apparatus are fully shown and described, in reference to the large wheel erected at Greenock for the Shaw's Water Spinning Co., and need not be here recapitulated; but it may be well to observe that a fault is often committed in making the governor not only bring the sluice geering into action, but to hold it in action until the motion of the wheel has been sufficiently increased or diminished by the elevation or depression of the sluice-plate. This is done for economy; and it is admitted that the apparatus is thus greatly simplified in its constructive details. But as all the pressure of throwing the geering out of action must be overcome by the centrifugal force of the pendulum balls, these must, therefore, of necessity be of great weight, and, consequently, less susceptible of being affected by slight variations of motion on account of their greater inertia. But the far greater evil consists in the geering being kept in action until the movement has been so far altered that the balls have acquired sufficient power to overcome the friction due to the pressure of the geering, and disengage itself; and which cannot possibly happen until the speed has been increased or diminished beyond the point required. To correct this the governor immediately falls into action to produce the contrary effect, which again is overdone, and must again be corrected. From this cause the movement of the wheel is never steady, but continually oscillates between two extremes, and the governing apparatus, though, in the first instance, less expensive, is speedily worn out, being constantly in action, and is never satisfactory. The principle to be kept in view is, to allow the pendulum balls to adjust themselves with perfect freedom to the velocity of the wheel, by giving them no other duty to perform; and, by a cam, obeying the motion of the balls, and shifting its position in obedience to theirs, to throw the geering into action in the manner done in the example above referred to, and which is fully described at pp. 76-78.

When the height of the fall is considerably less than the diameter of the wheel, we then apply the term *breast* as expressing the relation. The wheel depicted in Plates XCII.—XCIII., comes under this general denomination; and to denote that the water is received above the line passing horizontally through the axis, it takes the name of *high-breast.* These terms, however, are manifestly only relative; for if the wheel had been made of somewhat larger diameter, the water would have been thrown upon a lower point of the circumference, and changed the character of the wheel to that of *low-breast.* These terms, therefore, convey no other positive intimation respecting the size of the wheel than that its axis is below, above, or in the plane of the water level. It, however, usually suggests that the fall is low, and, consequently, that every precaution is taken to render it as much as possible available upon the wheel. For this purpose an *arc* is usually constructed of the same radius as that of the wheel, to confine the water and prevent it from being spilt from the buckets before it has arrived at the lowest point of the run. In the example referred to, this arc is built of hewn-stone, but sometimes it is constructed of timber, and not unfrequently of cast-iron plates. Sufficient clearance is, of course, necessary to allow of the wheel moving free of the arc, and to this extent there must always be a waste of water; but when the arc is properly constructed and wrought to the circle, this clearance need not exceed half an inch, which will be the measure of the plate of water which escapes without producing its effect upon the buckets immediately on their passing the plane of the axis. This arrangement is sometimes adopted with good effect in wheels belonging to the class called overshot, and it is applicable in all cases; but when the height of the fall is considerable, and the buckets properly constructed to retain and carry down the water, it is of less importance, as the small loss at the lower part of the revolution bears a diminished proportion to the whole effect realised. In the breast wheel the buckets are made more flat and radial than in wheels which receive the water near the summit, and are therefore not so well adapted to retain the water. The only reason assignable for this difference of form is, that it to some extent economises the fall; for in every wheel the lip of the bucket must descend below the water-level before any water can enter it, and consequently there is a loss of fall incurred in filling equal to something more than the depth of the bucket; and the lower the fall is this will evidently bear a greater ratio to the entire value of the water.

The power of all large wheels is taken off by a second shaft carrying a pinion, which geers with a large spur wheel bolted upon the shrouding of the water-wheel. This spur wheel is cast in parts or segments and bolted together, and is generally, though not invariably, an internal wheel. When there is considerable breadth between the shrouds, it is of importance to take the power at both sides, as shown in Plates XCII.—XCIII., and always from the loaded arc of the water-wheel. By this arrangement all strain is taken off the arms and journals of the wheel, which otherwise would be excessive. The circumstances also accommodate themselves to this arrangements, in so far as the centre of pressure of the mass falls usually within the depth of the shrouding, and renders any calculation unnecessary.

Wheels of a diameter up to 20 feet are frequently fitted with cast-iron arms, instead of being of the *spider* construction of that above referred to; and they have an advantage in being more rigid though more heavy, and therefore more severe on their journals. The common and best mode of fitting the cast-iron arms to the centres is shown by the annexed cut. The arms A A are cast with projecting cheeks on their lower extremities, which are planed and fitted into the dressed recesses of the centre, and are secured generally by three, but sometimes by one, bolt. The other extremities are cast with T-ends, which are likewise planed and fitted into corresponding recesses in the shrouds, and fastened by two bolts, and sometimes by adjustable keys.

Fig. 7.



It is not uncommon to find the arms of spider wheels secured with nuts instead of cotterals, as shown in the two examples given; but the latter mode is usually reckoned preferable, on account of the difficulty of maintaining the *truth* of the wheel in screwing up the nuts to the requisite degree of tightness; and the cotterals have the further advantage that they are more secure and less expensive. The constructive details of this mode of fitting are fully shown in Plate XCI., and require no further remark than that, with exception of the buckets, of which the curvature is not sufficiently great to retain the water long enough, the whole construction has our unqualified approval. The example to which these drawings refer is not only one of the very largest, but perhaps the most complete in its details, of any water-wheel yet constructed.

To determine the capacity of the wheel answering to a given supply of water, it is necessary to take into account the rapidity with which the buckets are filled and emptied—in other words, the angular velocity of the wheel. This velocity, it has been remarked, ought to be slightly less than that of the lamina of water falling into the wheel to prevent the back of the buckets, as they pass the receiving point, from striking against the descending stream, and thereby not only wasting a portion of the water by throwing it over the circumference of the wheel, but also diminishing the useful effect of that which passes into the buckets by the counteraction produced on the wheel.

The velocity of the stream may be generally determined in the following manner. Thus at the dam-sluice—if H be the depth to the centre of the sluice-opening, the velocity U of efflux will be expressed by

$$U = \sqrt{\frac{2gH}{1 + (\frac{1}{m} - 1)^2}};$$

m being a co-efficient depending for its value upon the particular form of the sluice-gate, but which may be taken generally $= 0{\cdot}64$; and g the symbol of gravity $= 32{\cdot}2$. Consequently, if we substitute the numerical values of these symbols, we shall have the simple arithmetical rule

$$U = 7{\cdot}133 \sqrt{H}.$$

And if the lead or course be short, and h the total fall from its origin to the extremity where the water is delivered upon the wheel, the velocity u at that point will be found from the formula

$$u = \sqrt{U^2 + 2gh},$$

in which U and g have the same significations as above.

But if the lead be of considerable length, it will be necessary to take into account the retardation which the water experiences in its passage. This is found from a calculation of the surface which the water passes over in its transit, and knowing approximately the velocity with which it moves. If we call u (found as above) the ultimate velocity of the stream, and U that at its origin, calculated at the dam sluice, then $\frac{1}{2}(u + U)$ will be nearly the mean velocity in the lead; and dividing the number of cubic feet of water to be delivered in a second by this mean velocity, the quotient will be the mean transverse area A of the current; and this divided by the mean width of the channel will give the depth of water. Now, the frictional retardation of a stream of water varying according to the amount of surface of the fluid in contact with the bed upon which it moves, is, therefore, inversely as the whole quantity of fluid—that is, for any given quantity of water, the resistance being as the surface of the bottom and sides of the channel directly, and as the whole quantity of water inversely, the diminution of velocity will be as $\frac{S\,L}{A}$, in which L is the length of the channel in feet, and S the surface (bottom and sides) over which the water glides. The retardation is expressed by

$$\sqrt{\left\{c\,\frac{S\,L}{A}\left(\frac{U+u}{2}\right)^2\right\}}$$

in which c is a coefficient determined by experiment $= .007$. If, therefore, U be the velocity due to the pressure at the sluice, and h be the total fall upon the channel the ultimate velocity will be expressed by

$$V = \sqrt{U^2 + 2gh - 0{\cdot}007\,\frac{S\,L}{A}\left(\frac{U+u}{2}\right)^2}.$$

It is scarcely necessary to remark that this is only an approximation, which, however, may be considered sufficiently correct for all practical purposes; and if greater exactness be desired, the value of V thus found may be substituted for u and the equation resolved anew for a nearer value of V.

As the value of h can be modified at pleasure in making the channel, it is of moment that these calculations be considered previous to determining the position of the shuttle. If the channel be already existing, and it is wished to determine the quantity of water flowing in it, this may be done with sufficient correctness, by finding the surface velocity, and multiplying it by the mean cross sectional area of stream: $\frac{4}{5}$ths of the quantity thus found will be very nearly the actual quantity flowing in the channel.

This rule, although often employed, and without material error when the quantity of water flowing is considerable and the velocity not great, is not to be relied upon when more than a rough approximation is desired. In some cases it is of importance to determine exactly the quantity of water supplied, as when a rental is paid for the power, when testing the efficiency of a wheel, or determining the power necessary to impell certain kinds of machinery; in these, and analogous cases, we must have recourse to more accurate formulæ. When the surface of the stream can be correctly ascertained over a portion of the channel, of which the cross section is nearly uniform, the following rule, which is founded on that of M. de Prony, may be employed with considerable confidence. Let U denote the mean velocity of the stream (which is sought to be determined), and V the surface velocity measured by a float in the middle of the stream, both reckoned in feet per second, then

$$U = \frac{V + 6.50}{V + 8.92} \times V.$$

As an example—let the surface velocity be 5 feet in a second, then will $V + 6.5 = 11.5$ and $V + 8.92 = 13.92$, and

$$\frac{V + 6.5}{V + 8.9} = \frac{11.5}{13.9} = .824;$$

therefore, $$\frac{V + 6.50}{V + 8.92} \times V = .824 \times 5 = 4.12 \text{ ft.}$$

that is, the surface velocity of the stream being 5 ft. per second, the mean velocity U is 4.12 ft. And this velocity being multiplied by the sectional area of the stream, the result will be the volume of water flowing in a second; and therefore,

$$Q = 60\,S \times U,$$

will be the quantity furnished in 60 seconds or 1 minute.

The maintaining power in a moving volume of water is obviously proportional to the quantity of descent in a given space; when, therefore, the motion is uniform, and is neither retarded nor accelerated by the force of gravitation, it is manifest that the whole weight of the water is employed in overcoming the frictional resistance offered by the bottom and the sides of the channel; and if the inclination vary, the relative weight, or the force which urges the particles along the inclined plane, will vary as the height of the plane when the length is given, or as the fall in a given distance. The retarding force, which is equal to the relative weight, must therefore also vary as the fall, and the velocity, which is as the square root of the impeding influence, must be as the square root of the fall; and supposing the hydraulic mean depth—that is, the depth which a current of water would take if spread out upon a horizontal surface equal to the bottom and sides of its channel—to be increased or diminished, the inclination remaining the same, the frictional resistance would be diminished or increased in the same ratio; and, therefore, in order to preserve its equality with the relative weight, it must be proportionally increased or diminished by increasing the square of the velocity in the ratio of the hydraulic mean depth, or the velocity in the ratio of its square root. We may, therefore, expect that the velocities will be conjointly as the square root of the hydraulic mean depth and of the fall in a given distance, or as a mean proportional between these two lines. From this reasoning, Eytelwein and some other writers on hydraulics, have deduced rules for determining the mean velocity of large bodies of water. Let δ denote the measure of declivity, and d the mean hydraulic depth of the current, then $100 \sqrt{\frac{\delta}{d}}$ will express the resulting velocity in feet per second—showing that, if the rate of descent were only one in 100 × 100, the stream would acquire a velocity represented simply by the square root of the hydraulic mean depth. If f denote the fall in feet each mile, the formula for the velocity will take the form

$$\frac{100}{\sqrt{5280}} \sqrt{\delta f} = \frac{11}{8} \sqrt{\delta f}.$$

Hence, the velocity reckoned in miles an hour is expressed by

$$\frac{11}{8} \cdot \frac{15}{22} \sqrt{\delta f} = \frac{15}{16} \sqrt{\delta f}.$$

The square root of the product of the hydraulic depth into the fall each mile in feet being diminished by $\frac{1}{16}$th, will hence represent the mean velocity of a river in miles each hour.

This rule, which is that deduced by Sir John Leslie for the velocity of water in rivers, may be compared with the result of Eytelwein's formula for the same purpose, as rendered by Dr. T. Young. Taking two English miles as a given length, he finds a mean proportional between the hydraulic mean depth and the fall in that space, and inquiring what relation this bears to the velocity in a particular case, finds that, in general, the mean proportional sought is $\frac{11}{10}$ths of the velocity in a second.

In order to test the accuracy of this rule, Dr. Young takes an example which could not have been known to Eytelwein. Mr. Watt observed, as Prof. Robison informs us in the article River, of the Encyclopedia Britannica, that in a canal 18 feet wide above, and 7 below, and 4 feet deep, having a fall of 4 inches in a mile, the velocity was 17 inches in a second at the surface, 14 in the middle, and 10 at the bottom; so that the mean velocity will be 14 inches or somewhat less in a second. Now, to find the hydraulic mean depth, we must divide the area of the section, $2\,(18 + 7) = 50$, by the breadth of the bottom and length of the sloping sides added together, whence we have $\frac{50}{20.6} \times 12 = 29.13$ inches; and the fall in two miles being 8 inches, we have $\sqrt{(8 \times 29.13)} = 15.26$ for the mean proportional of which $\frac{10}{11}$ is 13.87, the mean velocity in inches each second.

To test Sir John Leslie's rule by the same example, we have $f = \frac{1}{3}$ ft. and $\delta = \frac{29.13}{12} = 2.4275$ feet; therefore, $\delta f = .8092$ nearly, and $\sqrt{\delta f} = .8996$. Hence,

$$\frac{11}{8} \sqrt{\delta f} = \frac{11}{8} \times .8996 = 1.236 \text{ ft.}$$

that is, 14.83 inches, a result in excess of that found by Eytelwein's rule of 14.83 — 13:87 = 0.96 inch.

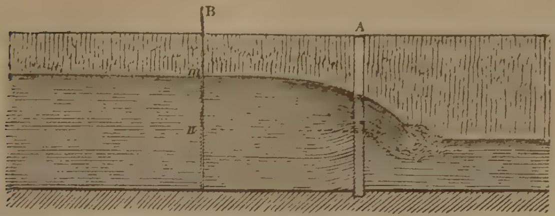
By M. de Prony's rule we have, $V = \frac{17}{12} = 1.4167$ ft., and

$$\frac{V + 6.50}{V + 8.92} = .77,$$

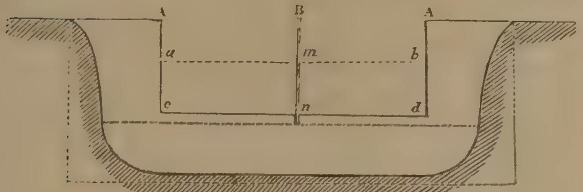
which, multiplied by 17, the surface velocity gives 13.1 inches as the mean velocity. And, taking the common rule of deducting a fifth from the surface velocity for the mean velocity of the current, we have $17 - \frac{17}{5} = 13.6$ inches, which does not differ greatly from M. Eytelwein's rule, in which we have most confidence, from our own experience, when the volume of water is very great. For smaller quantities of water, such as we find in leads cut to supply bucketted wheels, the modification which we have given of De Prony's rule is much more convenient, and we have found it, in general, very correct.

It is frequently difficult, and sometimes impossible, to apply any of these rules to determine the volume of water furnished; and it is often of importance, as when considerable accuracy is desired, to resort to more than one mode of measurement. In all ordinary experiments of this kind, there is a certain degree of uncertainty arising from inaccuracy of measurement; it is therefore of importance to have the means of checking the result of one rule by that obtained by a different process. The following method, which is not only simple and generally applicable, but likewise admitting of considerable accuracy, will therefore be useful. This consists in erecting a *notch* in some convenient part of the water course where the velocity is not great. The notch is easily formed in leads of moderate size by a board AA stretched across the channel, and having a rectangular part A*cd*A cut or notched out in the manner shown in the second of the annexed diagrams, and through which the whole of the water will be made to pass. The notch board being fixed, a rod B must be fixed vertically in the channel a few yards behind, and having a mark *n* upon it at exactly the level of the edge *c d* of the notch. The water being then permitted to descend in the lead, let its depth *n m* upon the rod B be carefully noted in inches, then taking from the second or third column of the annexed table the quantity corresponding to one inch of width at the depth noted, multiply that quantity by the whole width in inches, and the result will be the whole quantity flowing through the notch in cubic feet per minute.

Fig. 8.



Longitudinal Section of Lead.



Transverse Section of Lead.

Depth of the upper edge of the waste-board below the surface, in inches.	Cubic feet of water discharged in a minute by every inch of the notch, according to Du Buat's formula.	Cubic feet of water discharged in a minute by every inch of the waste-board of a weir, from experiments made by Mr. Smeaton.
1	0.403	0.428
2	1.140	1.211
3	2.095	2.226
4	3.225	3.427
5	4.507	4.789
6	5.925	6.295
7	7.466	7.933
8	9.122	9.692
9	10.884	11.564
10	12.748	13.535
11	14.707	15.632
12	16.758	17.805
13	18.895	20.076
14	21.117	22.437
15	23.419	24.883
16	25.800	27.413
17	28.258	30.024
18	30.786	32.710

Thus, if the depth from *m* to *n* on the rod B be 16 inches, and the width of the notch *a b* be 7 feet = 84 inches, then corresponding to 16 inches is 25.8 cubic feet in the second column, and which multiplied by 84 gives 2167.2 cubic feet as the whole quantity passing through the notch in a minute.

The quantity corresponding to 16 in the third column is 27.413 cubic feet, which multiplied by 84 gives 2202.7 cubic feet as the supply per minute, which is $35\frac{1}{2}$ cubic feet in excess of the result obtained by employing the second column. This discrepancy arises from the third column being calculated for weirs which discharge more water in a given time than notches, on account of their offering less impediment to the motion of the fluid. A weir is a wall built generally of solid masonry across the channel, with a parallel plank fixed horizontally on edge along the top of the building. The plank is termed the waste-board, and the water flows over it along the whole breadth of the channel, and thus suffers no lateral obstruction as it does in meeting the notch-board, in which the passage is usually contracted. But if the notch be made equal in width to the width of the channel, then this column ought to be employed, since under these circumstances the conditions are strictly analogous, and the notch may be called a weir.

When the preceding table cannot be conveniently applied, the value of Q, the quantity of water discharged in a minute, will be found very nearly from the expression,

$$Q = 200\ H\ L \sqrt{H},$$

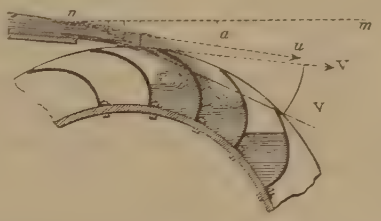
in which H is the height *m n* of the surface level of the water above the sole of the notch, and L the width, all in feet. Thus taking the example given above, we have $H = \frac{4}{3}$ feet, and L = 7 feet, therefore,

$$Q = 200 \times \frac{4}{3} \times 7 \sqrt{\frac{4}{3}} = 2155.3 \text{ cubic feet.}$$

To find the mean curve described by the lamina of fluid discharged upon the circumference of the wheel, when the water is carried over its summit, it is necessary, in the first place, to determine the velocity of the fluid vein at that point. Its rate of descent from the horizontal line *m n* in fig. 9. may then be assigned, by the following method.

Let u designate the velocity of the water at the extremity of the course, and a the angle, which the direction-board of the spout forms with the horizontal line *m n*, and which expresses the deflection of the line denoting the velocity u from the plane of the horizon; then the curve described by the sheet of water will be expressed by the equation

Fig. 9.



$$y = \frac{g\,x^2}{2\,u^2 \cos^2 a} + x \tan\ a,$$

x being the abscissæ measured upon the horizontal plane, taken at half the depth of the fluid vein, where the mean velocity is u; and y the vertical ordinates referred to the same initial point at *n*.

This equation may be expressed verbally thus:

To find the ordinates of the mean curve described by the water issuing upon a wheel from a shuttle, of which the direction board is inclined at a small angle with the horizontal plane, corresponding to any given horizontal abscissæ of the curve, multiply double the square of the velocity u of the water at the extremity of the direction board by the square of the cosine of the angle formed by its direction with the horizontal plane, and by this product divide the square of the given abscissa multiplied into $g = 32.2$. To the quotient which results, add the product of the same abscissa, multiplied into the tangent of the angle a of the velocity u with the plane *m n*. The sum will be the ordinate sought $= y$.

In giving any values to x equal to 3 inches, 6 inches, 9 inches, &c., we obtain the corresponding values of y, and a curve being traced through them, will give the path of the middle film of water.

When the direction board is horizontal, we have

$$a = 0, \cos a = 1, \tan a = 0,$$

and, therefore,

$$y = \frac{g\,x^2}{2\,u^2} = \frac{g}{2\,u^2}\,x^2,$$

which is the following verbal rule: divide 32.2 by double of the square of the velocity of the water at the extremity of the course, and multiply the quotient by the square of the given abscissa x. The product will be the ordinate y, corresponding to the value assigned to x.

If the water falls over a waste-board, as in breech and breast wheels, the mean velocity of the sheet will be obtained by taking $\frac{4}{5}$ths of that due to the whole height H of the level of the water above the edge of the board.

$$\text{Thus,} \quad u = \frac{4}{5}\sqrt{2\,g\,\text{H}} = 6.4\sqrt{\text{H}}.$$

This velocity will project it only slightly in a horizontal direction; and, if much accuracy is required, it is easy to determine for any such case the parabola described by the stream.

To exemplify the rule, as applied to the overshot wheel, we must, in the first place, determine the point of the circumference where the mean jet encounters the wheel, draw a tangent to the parabolic curve described by the fluid in the direction of its final velocity V; and, having done this, we calculate the height due to the velocity u, adding the distance of the point of contact below the origin of the curve. The velocity due to the sum of these heights, is that with which the water falls into the buckets.

Thus let it be required to find the final velocity of the water falling upon a wheel of $11\frac{1}{2}$ feet diameter, of which the axis is 10 inches before the vertical line, falling from the extremity of the direction board, which has an inclination of 1 in 12? If we suppose that the extremity of the direction board is .8 inch above the wheel, and that the mean depth of the lamina of water is 4 inches, and its velocity 9.84 feet in a second, then,

$$\tan a = \tfrac{1}{12} = 0.083, \ \cos a = 0.995, \ u = 9.84 \text{ ft.}$$

$$y = \frac{32.2\,x^2}{2\,(9.84 \times 0.995)^2} + 0.083\,x = 0.168\,x^2 + 0.083\,x.$$

Taking, therefore, any values of x at pleasure, we find the value of y by a simple arithmetical process. Thus, let $x =$ 3 in., we have $0.168\,x^2 = 0.168 \times 9 = 1.512$ and $0.083\,x = 0.747$; hence $y = 1.512 + 0.747 = 2.259$ inches.

The intersection of the curve thus determined with the circumference of the wheel is about $2\frac{3}{4}$ inches below the middle film of the vein at the extremity of the direction board; that is, from the origin of the curve, and the height due to the velocity, 9.84 feet a second being 1.52 feet, the total height due to the required velocity is 1 foot 9 inches, and, consequently, the velocity with which the water will strike the bottom of the buckets will be $\sqrt{64.4 \times 1.75} =$ 10.6 feet in a second, *less* the velocity of the wheel.

The lead or channel by which the water is supplied to the wheel ought obviously to be so constructed that it shall consume as little as possible of the available fall. As a first condition, it ought therefore to be as nearly straight as the local circumstances will admit; for at every bend which it makes, a portion of the impulse of the water will be absorbed by the concave side of the channel, and therefore a greater declivity will be necessary to bring forward a given quantity of water in the unit of time. Moreover, the centrifugal force created at the sinuosities has the effect of raising the surface and augmenting the abrasion of the banks at those points; and if by any accident a breach be produced, the sweep of the current must necessarily tend to enlarge the concavity with an accelerated progression. The inclination ought likewise to be as nearly as possible uniform—more correctly, it ought to be so regulated that the transverse sectional area of the stream shall remain constant throughout its whole length. If the course be constructed of masonry, it is obvious, from the remarks made respecting the effects of the hydraulic mean depth on the velocity of the current, that it will be of advantage to build the side walls vertically; and in order that the resistance may be the least possible, the depth of the stream should be equal to half the width. This rule may be followed when the quantity of water is small; but in leads of large area, the width at bottom is usually from four to six times the depth. When the depth is considerable the walls are moreover built with a certain amount of *batter*. Mr. Eytelwein, indeed, recommends that the breadth at the bottom be $\frac{2}{3}$rds of the depth, and at the surface $3\frac{1}{3}$. The area of such a section is twice the square of the depth, and the hydraulic mean depth $\frac{2}{3}$rds of the actual depth.

The slope here recommended is 4 to 3, forming an angle to the plane of the horizon with the water-way of 37°, whereas, in this country, the ratio commonly adopted for canals is 3 to 2, making an angle of 34°—which is far more than sufficient for any water-course intended merely for the purpose under consideration. The best angle to insure durability will, however, very much depend upon local circumstances, and the material of which the banks are constructed or composed.

Having fixed upon the dimensions of the lead and the intended depth of the current, if we call A the area of its transverse section, and q the volume of water to be brought forward in a second, the mean velocity U, which the water must have, will be expressed by $\text{U} = \frac{q}{\text{A}}$. And calling S the perimetrical surface (bottom and sides), and I the inclination or fall in 100 feet, which the channel ought to have to give the velocity U required, we shall have

$$\text{I} = \frac{\text{S}}{\text{A}}\,\text{U}\,(\cdot 00042\,\text{U} + \cdot 00444).$$

That is, to find the fall in a 100 feet of length of the channel, multiply ·00942 by the given velocity U; add to the product ·00444; multiply the sum again by the velocity and the product by the perimetrical surface, and divide the last product by the transverse sectional area of the channel.

Thus, if the quantity of water to be brought forward be 40 cubic feet every second, in a channel of 10 feet width, the depth of the water not to extend $2\frac{1}{2}$ feet, we shall have

A the area of the section = 10 × 2·5 ft. = 25 sq. ft.

U the mean velocity $= \frac{40}{25} = 1{\cdot}6$ ft. in a second.

S the perimetrical surface $= 10 + (2 \times 2\frac{1}{2}) = 15$ ft.; therefore, $\mathrm{I} = \frac{15}{25} \times 1.6\ (0.00942 \times 1.6 + .00444) =$ .0585 ft., the fall in 100 ft. of length of the channel.

This rule—at least the latter part of it, referring to the inclination—may be dispensed with when the channel is short. In cases where the whole run does not exceed a length of two or three hundred feet, the bottom may be made quite level, as the depth of the water in the channel will give sufficient fall for the velocity required, provided the area of section be made of ordinary magnitude.

Equal care is usually necessary in the construction of the tail-race as in that of the lead run; for it is of quite as much importance that the water leave the wheel pit freely as that it be brought forward with as little loss of fall as possible. The same rule will apply in both cases; but without some judgment in the engineer to apply it, with allowance for sinuosities, it is better to err on the side of excess in the case of the tail-race than to encounter the risk of flooding the wheel in backwater.

The quantity of water to be used being ascertained by the preceding methods, the capacity of the wheel may be readily determined. If q denote the volume of water flowing in a second of time, d the distance between two buckets reckoned upon the exterior circumference of the wheel, and v the velocity of those points of the circumference; it is evident that in one second, there will pass under the apron of the shuttle, a number of buckets equal to $\frac{v}{d}$; and consequently that each will receive a volume of water equal to q divided by $\frac{v}{d}$ that is $= q\frac{d}{v}$. But it is manifestly necessary that the bucket be capable of containing not only this quantity, but even a quantity about three times as great, otherwise a portion of the water will be spilt from the buckets too soon, and without producing its effect upon the wheel. If l represent the width of the bucket, that is, the width of the wheel within the shrouds, and s the area of its transverse section—more correctly the area of the section of the mass of fluid which it contains at the moment it passes the jet, $s\,l$ will represent its capacity, and in relation to the section of the bucket itself we shall have

$$s\,l = 3\,q\frac{d}{v} = 180\ \frac{q}{\mathrm{M\,N}} = 3\ \frac{\mathrm{Q}}{\mathrm{M\,N}};$$

M being the number of buckets in the wheel, and N the number of revolutions which the wheel makes in a minute. And since

$$d = \frac{\pi\,\mathrm{D}}{\mathrm{M}} \text{ and } v = \frac{\pi\,\mathrm{D\,N}}{60}. \text{ therefore } l = \frac{3\,\mathrm{Q}}{\mathrm{M\,N}\,s},$$

which is the width of the wheel when Q is the volume of water to be employed upon it in a minute, and when it is expected to realize the maximum mechanical effect of the water.

We proceed to establish the values of these symbols from considerations involved in the *modus operandi* of the wheel, but for practical purposes we may remark that with slight variation s may be taken $= \frac{3}{4}$ square foot; $l = 4.5\ \frac{\mathrm{Q}}{\mathrm{N\,D}}$ and $\mathrm{M} = 2.88\ \mathrm{D}$.

We have already seen that the whole dynamical force of the stream of water employed in impelling a wheel of any form is expressed by W H; but as the whole height H can in no case be rendered effective, we have found it necessary to affect this product by a co-efficient m which is always a proper fraction expressing the ratio of the force expended to that realized by the particular mover. To ascertain the theoretical value of m, which often, however, differs considerably from the actual value; let us take, in the first place, the overshot wheel as the simplest case which the problem presents. Referring to fig. 1, p. 213,—let the horizontal lines M A and F B represent the higher and lower water levels; the vertical distance A B will then indicate the entire height H of the fall. If we divide this height into three parts: A c the part comprised between the higher surface of the water and the point where the stream strikes the wheel; c D equal to the height of the arc of the wheel, which may be considered as filled with water; and B D, the distance between the point at which the water may be considered to be wholly discharged and the bottom of the fall; this division will enable us to particularize and estimate the losses which take place between the several partial limits, and therefore between the extreme limits A and B.

Within the limit c and D we have the whole effect of the expenditure realized upon the wheel, since the whole volume of fluid acts constantly, and with its whole weight in the vertical direction, it must realize upon this part of the wheel an effect corresponding to the height through which it descends, and yield a result which, in conformity with the notation adopted, will be expressed by W × c D. The height A c is that due to the velocity with which the water falls into the buckets, and with which it would strike the start-boards, if it did not experience any diminution between the shuttle and the point of impulse. But this condition does not hold true; for at the point where the fluid encounters the wheel, the height measuring the velocity is not $= \mathrm{A}c = h$, for brevity, but to a quantity $= h - \mu\,h$. The value of the co-efficient μ will depend upon various conditions: In the first place, there arises a loss equivalent to a loss of velocity from the contraction of the fluid vein *(vena contracta)* which the fluid experiences in its passage through the orifice of the shuttle; secondly, from the resistance offered by the surfaces over which it passes; thirdly, from the dispersion of the filaments of the fluid by striking against the oblique plates of the buckets; and, fourthly, from the oblique direction with which the mean volume of fluid arrives at the bottom of the buckets. This obliquity may, in general, be taken at 30°, causing a diminution in the value of h of 0.14, and consequently a corresponding diminution of the force of the impulse. All these causes combined are found according to local circumstances, such as a good or bad arrangement of the shuttle and direction plate or apron, to be equivalent to from $\frac{2}{10}$ to $\frac{3}{10}$ of the whole value of h. Let A a be the portion of the height A c, representing the value of $\mu\,h$, the remaining part $a\,c$ will then represent the height $h - \mu h$, due to the actual velocity V of the jet, and consequently equal to $.0155\ V^2$.

From what has been before explained (p. 210) regarding the impulsive action of a current of water, this height will be subject to two other reductions: the one $h' = .0155\ v^2$ is that due to the velocity v of the wheel in feet per second, and therefore increases with that velocity; the other $h'' = .0155\ (V - v)^2$ is the height due to the velocity lost by the shock, and which, on the contrary, decreases as the velocity v increases. The sum of these losses will be the least possible, or $.0155\{v^2 + (V - v)^2\}$ will be a *minimum* when $v = \frac{1}{2}\ V$. They will be respectively equal, that is, each will be $\frac{1}{4} \times .0155\ V^2 = \frac{1}{4}\ h\ (1 - \mu)$; and the two together that is $a\,b + b\,d$ will be equal to $\frac{1}{2}\ h\ (1 - \mu)$. In this case of *minimum* loss, the remaining part $d\,c$, which is all of the fall A c that can be regarded as effective, will therefore be equal also to $\frac{1}{2}\ h\ (1 - \mu) = \frac{1}{2}\ a\,c$, and consequently less than $\frac{1}{2}\ h = \frac{1}{2}$ A c, that is, than half the head reserved between the surface level of the water and the point at which it is received upon the wheel.

Although the sum of the two losses h' and h'' cannot thus be less than $\frac{1}{2}\ h\ (1 - \mu)$, we know that it can be, and is, indeed, almost always considerably greater, and increases as the difference between h' and h'' increases. It will attain its *maximum*, if one of the two quantities, h'' for example, should become zero, giving rise to the condition $V = v$. In that case we have $h' = .0155\ V^2 = h\ (1 - \mu)$, that is $a\,b = a\,c$, showing that no part of the fall A c remains effective. But in practice this condition can never arise unless by miscalculation of the primary values of V and v; this last must always be less than the former, and, consequently, h'' must always have a real value; and in every case where h'' is greater than zero, it is manifest from what has gone before, that a certain portion, however small, of the height A c must remain effective. Theoretically, this is shown to be less than $\frac{1}{2}h$, and under the very best arrangements it cannot in practice be expected to amount to $\frac{2}{5}h$, and in ordinary cases it ought not to be assumed greater than $\frac{1}{3}h$. As a general rule in practice it may therefore be admitted, that in bucket wheels about two-thirds of the part of the fall comprised between the level of the water at the shuttle, and the point where the fluid encounters the wheel, is lost as respects the effect produced. The actual value of this height h, which may be generally expressed by $W\ \{h\ (1 - \mu) - h' - h''\}$ will therefore be represented by

$$W\ (h - \tfrac{2}{3}\ h) = \tfrac{1}{3}\ W\ h,$$

a result which we shall subsequently find is closely analogous to that obtained experimentally as the effect of the best forms of impulsive wheels.

Since then a third only of the part of the fall above the wheel is available as power, whilst the whole of the part from that point downwards to the turn of the buckets, namely, c D, the height of the loaded arc, is entirely realized, it is manifestly of advantage to augment this latter part as much as possible at the expense of the former. But this augmentation has a near limit, since there would be no economy, but the converse, in raising the point of reception so much that the water in arriving at the wheel would have a less velocity than the buckets. In this case it could not begin to act upon the wheel, except as a retarding influence, until, by an acceleration of its velocity, it established the necessary condition $V > v$.

The portion of the fall D B, from the bottom of the loaded arc downwards, is evidently lost without answering any beneficial purpose. This loss arises from two causes—rather consists of two parts—the part e B is a measure of the loss resulting from the form of the buckets, and the small portion D e of that caused by the velocity of the wheel, or more correctly, it is a measure of the loss occasioned by the centrifugal force produced in the fluid by the angular velocity communicated to it in its descent in the buckets. Leaving this effect out of view in the meantime, it is evident that, if not influenced by any central force, the surface of the water contained in the buckets would continue horizontal. In proportion as the buckets descend, in consequence of the revolution of the wheel, this surface gradually approaches the lip of the containing or front plate of the bucket; and the instant after it arrives at this position, marked $h\,i$, it begins to be discharged, and the bucket will be completely emptied when the face has attained the horizontal position marked $k\,l$. The arc F h, which measures the distance of the base of the wheel to the point where the water begins to be discharged, will therefore include the partially loaded arc $h\,k$ and the empty arc k F. This last is equal to the angle $u\,k\,l$, which the front plate of the bucket makes with the tangent to the circumference, an angle which is known from the rules employed in tracing the lines of the buckets, and which we may here designate by a. The arc F h = F $k + k\,h$, and this last $k\,h$ is equal to the angle $x\,h\,i$, which the front plate of the bucket makes with the surface of the water at the point where the fluid begins to be discharged, and which for brevity may be called z. We have, therefore, F $h = a + z$.

Whatever may be the magnitude of the two arcs of discharge, from these data we can always determine the rate of diminution of the quantity of water remaining in the buckets between the extreme points where it begins to overflow and where it leaves the bucket entirely empty, and therefore can determine the *mean arc of discharge*, and from this the mean quantity of effect due to the water carried below the commencement of the arc at h, in terms of the whole quantity which the buckets would be capable of carrying in their horizontal position in passing through the height c D. By determining the mean of this arc, we find the point at which, if the whole water were instantly discharged from each bucket as it passed, the effect upon the wheel would be the same as takes place when it is prolonged over the whole length of the arc $h\,k$. Generally[*] this may be taken as the arithmetical mean distance between h and k, and may be expressed by $a + \frac{1}{2}\ z$. If upon A B we take e at such a height that a horizontal line meeting the circumference of the wheel at a point equidistant from h and k, then will e B be the total loss of fall arising from the reversion of the buckets; and the wheel will yield the same result as if the entire water, instead of being gradually discharged between these points, were carried down to the point e, and then instantly thrown from the buckets.

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[*] Excepting, however, those cases in which an arc is employed to retain the water in the buckets of the wheel—a condition which will be subsequently adverted to (p. 235).

The arc below that point may, therefore, be regarded as entirely empty and producing no effect; and to designate its relation we have e B equal to the versed sine of that mean arc, of which the semi-diameter of the wheel is the radius; and, therefore, putting D to denote the diameter, we shall have

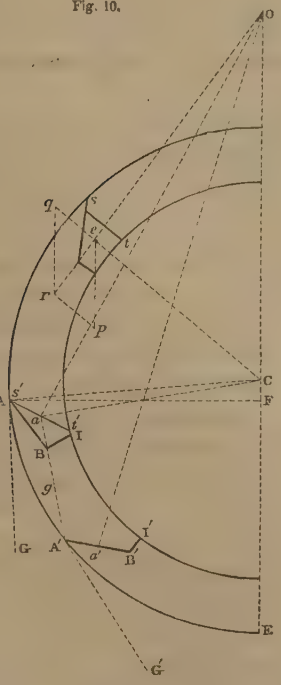
$$e\,\mathrm{B} = \tfrac{1}{2}\,\mathrm{D}\,\{1 - \cos\,(a + \tfrac{1}{2}\,z)\}.$$

It may be remarked that the angle z, which the surface of the fluid makes at the commencement of the discharge with the front plate of the bucket, will depend upon the volume of water in the buckets, as well as upon the form and dimensions of these, both of which are, of course, either known from the rules employed in the design of the wheel, or may be ascertained by direct measurement.

It remains to determine the loss of head resulting from the centrifugal force produced in the fluid filling the buckets, by the motion of the wheel. This loss is sometimes considerable, although not commonly reckoned among the influences to which wheels of this class are liable. M. Poncelet was the first, we believe, to direct attention to it, and has established a theorem for its determination, which may be said to complete the theory of the *modus operandi* of bucketted wheels.

It has been already stated (page 211) that if a body move in a circle at a distance x from the centre, its centrifugal force will be expressed by $\frac{w}{g}\,\omega^2\,x = \frac{w}{g}.\frac{v^2}{x}$, when $\frac{v}{x}$ is put for ω, the angular velocity with which the body revolves.

Fig. 10.



Now since every molecule of fluid contained in the buckets of a wheel in motion is subjected to the action of the two forces—that of gravity and the centrifugal action, we may confine our attention to one such molecule e of which the mass $\frac{w}{g}$ may, for brevity of expression, be called m. If $e\,p$, in the annexed diagram, represent the force $m\,g$ of gravity acting vertically, and $e\,q$, measured in the direction of the radius C e represent the centrifugal force $m\,\omega^2\,x$, the diagonal $e\,r$ of the parallelogram, will be the resultant of the two forces, and may be regarded as representing the measure and direction of a new force replacing the two actual forces $e\,p$ and $e\,q$, and producing upon the molecule the same intensity of action. If we prolong $e\,r$ until it meets the vertical line E O passing through the centre C in O, this point will be such that C O $= \frac{g}{\omega^2}$, and, therefore, is independent of the position of the molecules, and the same for all—all the resultants of the forces converging to that point, which is therefore the centre of action whence all the forces are directed. The surface of the fluid being always perpendicular to the direction of the force which acts upon the molecules, that of the fluid contained in all the buckets will be so to the lines passing to the point O from any point of the wheel, and, consequently, the section $s\,t$ of any given surface will be an arc of a circle having O for its centre.

In the revolution of the wheel, the extremity s of this arc approaches gradually the lip of the front plate of the bucket, and will arrive at it whenever the bucket shall have come into the position A B I. Immediately after it will begin to be discharged, and the discharge will continue until the bucket has descended to the position A′ B′ I′, where the limiting arc of the fluid surface will have passed under the bucket plate A′ B′.

In wheels moving at ordinary velocities the surface of the water in the buckets may be regarded as planes perpendicular to lines drawn to them from the centre O. On this supposition the two arcs of discharge, A E and A′ E, may be thus determined. The first, that is, the whole arc measured by the angle A C E, is equal to

$$\mathrm{G\,A\,F} = \mathrm{G\,A\,B} + \mathrm{B\,A}\,t' + t'\,\mathrm{A\,F} = a + z + y,$$

in designating the angle t' A F $= a$ O C by y, the point a being equidistant between s' and t'. Taking $a\,g$ perpendicular to a C, and calling b the angle which the first of these lines makes with the tangent A G, supposing both produced until they meet, and which will necessarily be equal to the angle A C a; and

$$\angle\,\mathrm{O}\,a\,\mathrm{C} = \angle\,g\,a\,t' = \angle\,\mathrm{G\,A}\,t' - b.$$

Moreover, the triangle O a C gives

$$\sin\,a\,\mathrm{O\,C} : a\,\mathrm{C} :: \sin\,\mathrm{O}\,a\,\mathrm{C} : \mathrm{O\,C},$$

that is, $$\sin\,y \quad : \rho \quad :: \sin\,(a + z - b) : \frac{g\,r^2}{v^2},$$

in which r is the dynamical radius of the wheel, and ρ the same radius, diminished by half the depth of the buckets. From this analogy we obtain

$$\sin\;y = \frac{\rho\,v^2\,\sin\,(a + z - b)}{g\,r^2}.$$

The angle b being generally very small will very slightly influence the value of y, and may therefore be neglected in determining the angle of discharge, without sensible error; but by way of compensation, we may substitute for ρ the dynamical radius r making

$$\sin\;y = \frac{v^2}{gr}\,\sin\,(a + z).$$

To find the measure of the arc A′ E: if we take y' to denote the angle a' O E we shall have A′ E = $\angle\,(a + y')$. And from the same species of reasoning employed for y we find

$$\sin y' = \frac{v^2}{gr}.\ \sin a.$$

From these values we might establish a general expression for the mean arc of discharge; but, as already remarked, this does not differ sensibly from the arithmetical mean, and may therefore be represented by $a + \frac{1}{2}z + \frac{1}{2}y + \frac{1}{2}y'$. And reverting to Fig. 1, its versed sine will be represented by D B, which is the whole loss of head resulting from the form of the buckets and the centrifugal force conjoinedly, and may thus be calculated previous to the construction of the wheel. If we designate this loss by h''' we shall have

$$h''' = \tfrac{1}{2}\,\mathrm{D}\,\{1 - \cos\,(a + \tfrac{1}{2}z + \tfrac{1}{2}y + \tfrac{1}{2}y')\}.$$

As an example of the arithmetical process of determining this loss of head from *a priori* data, let the diameter of the wheel be 37½ feet; the number of buckets 92; the width l = 3.55 feet = 3 feet 6½ inches; and the depth 12.8 inches. The breadth AB of the first plate of the buckets is 1.522 feet, and a line joining AI = 1.683 feet. The angle GAB which AB makes with the tangent AG to the circumference is 31° 37′ = a; BAI = 9° 8′; and AIB = 53° 10′; the surface of the triangle BAIB is therefore = .20344 square feet = s'. The dynamical radius r of the wheel is 17.935 feet, and the distance d between the buckets measured on the circle described by that radius is 1.225 feet; the velocity v at the same circle = 8.2 feet in a second. The quantity of water furnished in the same unit of time is 5.3 cubic feet = q. The section s of water contained in a bucket before it begins to discharge will therefore be

$$s = \frac{qd}{lv} = \frac{5.3 \times 1.225}{3.55 \times 8.2} = 0.22299 \text{ sq. ft.}$$

But this section being greater than .20344 = s' shows that the water surface meets the sole at some point t' higher than I. Now the angle z, that is, the angle which the surface of the water makes with the face plate of the bucket will be equal to the sum of the angles BAI and IAt'.

To find this last we may take without sensible error,*

$$\tan \mathrm{IA}t' = \frac{2\,(s - s')}{\mathrm{AI}^2 - 2\,(s - s')\tan \mathrm{ABI}}$$

$$= \frac{.0391}{2\ 8325 - .006286} = \tan 0° 47' 34'';$$

therefore, $z = \mathrm{ABI} + \mathrm{IA}t' = 9° 8' + 0° 47'\ 34 = 9° 56'$ nearly.

To find y and y' we have, for the first,

$$\sin y = \frac{v^2}{gr}\sin\,(a + z) = \frac{(8.2)^2}{32.2 \times 17.935} \times \sin\,(31° 37' + 9° 56')$$

$$= \sin 4° 26' \text{ very nearly.}$$

$$\sin y' = \frac{v^2}{gr}.\ \sin a = \frac{(8.2)^2}{32.2 \times 17.935} \times \sin 31° 37'$$

$$= \sin 3° 30' \text{ very nearly.}$$

For the whole ineffective arc of discharge we have therefore

$$a + \tfrac{1}{2}z + \tfrac{1}{2}y + \tfrac{1}{2}y' = 40° 33';$$

and from this we obtain as the total loss

$$h''' = \tfrac{1}{2} \times 37\tfrac{1}{2}\,(1 - \cos 40° 33') = 4.502 \text{ feet.}$$

If we decompose this into the losses incurred by the form of the buckets and the centrifugal force, we find for the former

$$c\mathrm{B} = \tfrac{1}{2} \times 37\tfrac{1}{2}\,(1 - \cos 36° 35') = 3.694 \text{ feet};$$

$$\text{and} \quad \therefore c\mathrm{D} = (4.502 - 3.694) \text{ feet} = 0.808 \text{ feet.}$$

These parts are, therefore, very nearly as 100 to 22; consequently, retaining these numbers, the loss of fall below the loaded arc is shown to be increased by the centrifugal force communicated to the fluid from 100 to 122.

In conformity with the principle before indicated we must, in order to arrive at a complete theoretical expression of the value of the fall, subtract these several losses, and multiply the remainder by the weight of water for the total effect, which will then be expressed by

$$\mathrm{W}\,(\mathrm{H} - \mu h - h' - h'' - h''').$$

But this expression being deduced entirely from theoretical considerations, we must, in order to compare it with the results of experience, applicable to every particular case, introduce our co-efficient of reduction m, when we will have as the actual effect developed by the wheel

$$\mathrm{E} = m\mathrm{W}\,(\mathrm{H} - \mu h - h' - h'' - h''').$$

From this it therefore appears that in every bucket wheel the ultimate effect will be increased as the five quantities μ, h, h', h'', h''' are diminished. Now these have respect,

μ to the construction of the shuttle and water-course, which ought accordingly to be adapted with care to the particular case;

h to the diameter of the wheel which, therefore, ought to be as great as the other conditions will admit, (it being understood here that the wheel is constructed on the overshot principle);

h' and h'' to the difference of velocity between the water and the wheel for a given value of h; a condition which will be satisfied the more nearly as the velocity of the wheel approaches half the velocity of the water at the moment it arrives at the bottom of the buckets;

h''' to the proper disposition and form of the buckets, and a small velocity of the wheel, by which the water will be carried to the lowest point possible of the fall before it is discharged.

The most trustworthy experiments on wheels of this class published in this country are those of Mr. Smeaton, made in 1759, upon a small model wheel of 2 feet diameter. Various details are, however, wanting to enable us to compare his results with the preceding formulæ—especially the form and dimensions of the buckets. The following table contains the summary of his results.

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* When the point t' falls below I, that is, when s' is greater than s, the formula for z becomes

$$\tan z = \frac{2s\,.\,\mathrm{AB}}{\mathrm{AB}^2 + 2s \cot\,(180° - \mathrm{ABI})}.$$

No.	Whole descent.	Water expended in a minute.	Turns at the maximum in a minute.	Weight raised at the maximum.	Power of the whole descent.	Power of the wheel.	Effect.	Ratio of the whole power and effect.	Ratio of power of the wheel and effect.	Mean Ratio.
	Inch.	*lb.*		*lb.*						
1	27	30	19	6½	810	720	556	10 : 6.9	10 : 7.7	Medium 10:8.1
2	27	56⅔	16¼	14½	1530	1360	1060	10 : 6.9	10 : 7.8	
3	27	56⅔	20¾	12½	1530	1360	1167	10 : 7.6	10 : 8.4	
4	27	63⅓	20½	13½	1710	1524	1245	10 : 7.3	10 : 8.2	
5	27	76⅔	21½	15½	2070	1840	1500	10 : 7.3	10 : 8.2	
6	28½	73⅓	18¾	17½	2090	1764	1476	10 : 7.	10 : 8.4	10:8.2
7	28½	96⅔	20¼	20½	2755	2320	1868	10 : 6.8	10 : 8.	
8	30	90	20	19½	2700	2160	1755	10 : 6.5	10 : 8.1	10 : 8.2
9	30	96⅔	20¾	20½	2900	2320	1914	10 : 6.6	10 : 8.2	
10	30	113⅓	21	23½	3400	2720	2221	10 : 6.5	10 : 8.2	
11	33	56⅔	20¼	13½	1870	1360	1230	10 : 6.6	10 : 9.	10 : 8.5
12	33	106⅔	22¼	21½	3520	2560	2153	10 : 6.1	10 : 8.4	
13	33	146⅔	23	27½	4840	3520	2846	10 : 5.9	10 : 8.1	
14	35	65	19¾	16½	2275	1560	1466	10 : 6.5	10 : 9.4	10 : 8.5
15	35	120	21½	25½	4200	2880	2467	10 : 5.9	10 : 8.6	
16	35	163½	25	26½	5728	3924	2981	10 : 5.2	10 : 7.6	
1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.

From this table we perceive the small effects produced by an increase of the head $A c = h$ above the wheel. On the general results, he observes, that "the power of the water computed from the height of the wheel only, appears to observe a more constant ratio" than that between the power of the water reckoned from the whole descent and the ultimate effect. Thus the ratios in column 9 differ from that of 10 : 7.6 to 10 : 5.2 ; whereas, taking the mean set down in col. 11, "we find the extremes to differ no more than from the ratio of 10 : 8.1 to 10 : 8.5 ; and as the second term of the ratio gradually increases from 8.1 to 8.5, by an increase of head from 3 inches to 11 inches, the excess of 8.5 above 8.1 is to be imputed to the superior impulse of the water at the head of 11 inches above that of 3 inches ; so that, if we reduce 8.1 to 8, on account of the impulse of the 3-inch head, we shall have the ratio of the power, computed upon the height of the wheel only, to the effect at a maximum as 10 : 8, or as 5 to 4 nearly ; and from the equality of the ratio between the power and effect, subsisting when the constructions are similar, we must infer that the effects, as well as the powers, are as the quantities of water and the perpendicular heights multiplied together respectively."

These inferences are corroborative of the principles which we have attempted more formally to illustrate ; but we must also quote his remarks "concerning the velocity of the circumference of the wheel, in order to produce the greatest effect," as they are still frequently appealed to in justification of an erroneous interpretation of a true doctrine. The doctrine is thus stated by the author:—"If a body is let fall freely from the surface of the head to the bottom of the descent, it will take a certain time in falling ; and in this case the whole action of gravity is spent in giving the body a certain velocity : but if this body in falling is made to act upon some other body, so as to produce a mechanical effect, the falling body will be retarded ; because a part of the action of gravity is then spent in producing the effect, and the remainder only giving motion to the falling body : and therefore *the slower a body descends, the greater will be the portion of the action of gravity applicable to the producing a mechanical effect ;* and in consequence the greater that effect may be.

"If a stream of water falls into the bucket of an overshot wheel, it is there retained until the wheel by moving round discharges it : of consequence the slower the wheel moves, the more water each bucket will receive ; so that what is lost in speed, is gained by the pressure of a greater quantity of water acting in the buckets at once ; and, if considered only in this light, the mechanical power of an overshot wheel to produce effects will be equal whether it moves quick or slow : but if we attend to what has been just now observed of the falling body, it will appear that so much of the action of gravity, as is employed in giving the wheel and water therein a greater velocity, must be subtracted from its pressure upon the buckets, so that, though the product made by multiplying the number of cubic inches of water acting in the wheel at once by its velocity will be the same in all cases ; yet as each cubic inch, when the velocity is *greater* does not press so much upon the bucket as when it is *less*, the power of the water to produce effects will be greater in the less velocity than in the greater : and hence we are led to this general rule, *that* cæteris paribus, *the less the velocity of the wheel, the greater will be the effect thereof.*"

According to this view of the subject we ought to introduce into our formula a further reduction of H depending upon the velocity of revolution, and which would therefore be a function of v. But if the mode in which h'' has been obtained be observed, it will be found that the circumstance which Mr. Smeaton had in view is there included. It is admitted that the gravitation of the fluid in the buckets cannot at the same time be producing pressure and velocity ; but we have laid it down as a condition, which Mr. Smeaton also insists upon, that the water must have a higher velocity than the circumference of the wheel at the moment of its passing into the buckets. This condition being fulfilled, it is then clear that as no additional velocity has been generated in the fluid, after it has entered the buckets, no part of its power is thereby consumed below that level, and that all its effect will be realized upon the wheel. In other words, reverting to Fig. 1, page 213, the effect of the volume of water on the loaded arc will be expressed by $W \times c D$.

This may be exhibited somewhat more formally, and as a preliminary step let it be required to prove that the weight of fluid carried in the loaded arc of the wheel, from the level of c to the lower level D, is equal to the effort which would be exercised by the weight of a prism of water G H placed at the extremity of the dynamical radius O P, the height of the prism being equal to c D, and the area of its base equal to the cross section of the fluid arc, if the water in the buckets were uniformly distributed, and formed a continuous arc. To show that this is true statically, it will be sufficient to prove that the moments of pressure are in the two cases equal. For this purpose let it be supposed that the length of the fluid arc is divided into an infinite number of small elementary arcs, such as $m\,n$ having a cross section σ. If then we designate by ϕ the specific gravity of the fluid, we shall have $\sigma \,.\, m\,n \,.\, \phi$ as the weight of the small arc $m\,n$. And since it acts vertically, the distance between the direction of its pressure and the centre of rotation will be the horizontal line $r\,s$. Thus then we have as the moment of its pressure $\sigma \,.\, m\,n \,.\, \phi \,.\, r\,s$. But the triangles $m\,n\,t$ and $r\,\mathrm{O}\,s$ are similar; hence $m\,n \,.\, r\,s = \mathrm{O}\,r \,.\, p\,q$, and therefore,

$$\sigma \,.\, m\,n \,.\, \phi \,.\, r\,s = \sigma \,.\, \phi \,.\, \mathrm{O}\,r \,.\, p\,q.$$

Now the sum of all the partial moments will be the moment of the entire arc, and will be found by multiplying the common factor $\sigma \,.\, \phi \,.\, \mathrm{O}\,r$ by the sum of all the small heights $p\,q$ of the elementary arcs; this sum is evidently $f\,g = c\,\mathrm{D}$; the entire moment will therefore be $\sigma \,.\, \phi \,.\, \mathrm{O}\,r \,.\, c\,\mathrm{D}$. But that of the prism G H is manifestly $= \sigma \,.\, \phi\, \mathrm{G\,H} \,.\, \mathrm{O\,P}$; and since $\mathrm{G\,H} = c\,\mathrm{D}$ and $\mathrm{O}\,r = \mathrm{O\,P}$, the two moments are equal.

If we now bring into view the dynamical conditions imposed by the motion of the wheel, and keep in view that no motion is *communicated* to the water within the limits of the arc, we have as data a pressure applied at P in the direction of movement $\sigma \,.\, \phi \,.\, c\,\mathrm{D}$, and a velocity at that point of v feet per second: the force impressed will therefore be expressed by $\sigma \,.\, \phi \,.\, c\,\mathrm{D} \,.\, v$. And if q be the volume of water flowing in a second, with a continuous section σ, and velocity v communicated independently of the motion of the wheel, and acquired before it reached it, we have $q = \sigma \,.\, v$; and w being the weight of the volume q, we have besides $w = \phi \,.\, q$. Taking the values of σ and ϕ in these two equalities, and substituting them in the expression above of the force impressed, it becomes

$$\frac{q}{v} \,.\, \frac{w}{q} \,.\, c\,\mathrm{D} \,.\, v = w \,.\, c\,\mathrm{D},$$

which is the condition we undertook to demonstrate, and upon which it will be observed, the velocity of the wheel has no influence.

The best data which we possess for determining the value of the co-efficient m in our formula of the actual efficiency of the wheel is a table of experiments furnished by M. D'Aubuisson containing all the conditions. The mean of these cases gives $m = .8997$, and the highest value is .917, and the lowest .874. We may, therefore, without serious error put $m = .9$. The other terms may also be simplified for the purposes of ready application. Thus the three terms μh, h', h'' taken together, we have already shown, do not differ widely from $\frac{2}{3}\,h$; and, except in extreme cases, h''' will not vary more than between $\frac{1}{7}$ and $\frac{1}{5}$ of the diameter of the wheel. Let us assume the mean of these extremes, namely, $\frac{1}{6}$ D and substitute these quantities with a value of $m = 0.9$ in our formula, it will then be reduced to the following:

$$\mathrm{E} = 0.9\ \mathrm{W}\,(\mathrm{H} - \tfrac{2}{3}h - \tfrac{1}{6}\mathrm{D}),$$

and by putting Q = the number of cubic feet of water furnished in a minute, and expressing the effect in units of horse-power, conformably to the principle before explained (p. 207), we have

$$\mathrm{E} = .0017\ \mathrm{Q}\,(\mathrm{H} - \tfrac{2}{3}h - \tfrac{1}{6}\mathrm{D}).$$

As an example, let the quantity of water be 1000 cubic feet a minute, and the fall 25 feet; in this case the wheel would be made about 22 feet diameter; therefore $\frac{2}{3}h = 2$ feet and $\frac{1}{6}\mathrm{D} = 3\frac{2}{3}$ feet. Hence, the value of the fall would be reduced to $25 - (2 + 3\frac{2}{3}) = 19\frac{1}{3}$ feet, and this multiplied by $1000 = 19333$. Finally, $19333 \times .0017 = 32.87$ horse-power.

This formula may also be employed to determine the volume of water which it would be necessary to employ, with a given head, to obtain any required amount of power—a problem which very frequently occurs in practice.

In this we have confined our attention to the case of the overshot wheel, on account of its being the most obvious; but the same formula may, by a very slight modification, be applied to determine the effect of a breech-wheel. The modification referred to is simply the replacing of $\frac{1}{6}$D by its assumed equivalent h''', by which we have

$$\mathrm{E} = .0017\ \mathrm{Q}\,(\mathrm{H} - \tfrac{2}{3}h - h''').$$

We replace h''' because in wheels of this kind its value ought to be always less than in the overshot arrangements. The breech-wheel, as we have already seen, has usually a diameter somewhat greater than the height of the fall; and as h''' is proportional to the diameter, we have by this arrangement the advantage of making it as small as possible within the limits of practice. We can, indeed, increase the diameter at pleasure, and thereby proportionally increase the length of the loaded arc—the grand source of power in the bucketed-wheel of whatever form.

Tredgold, in his edition of R. Buchanan's Essays, has endeavoured to fix the proportion of the radius of the wheel to the height of fall to yield a maximum effect, but seems to have left out of view the effect of the centrifugal force, and to have supposed the wheel to revolve in an arc—which is, indeed, the usual arrangement now adopted. The following is his mode of calculation:—Let c = that portion of the circumference which is to be loaded with water—that is, the portion of the half circumference below the point at which the water flows upon the wheel; and let x = the arc comprehended between that point and the horizontal plane passing through the axis of the wheel; also make b = the area of the stream supplying the buckets. Then the solid which represents the effective force, that is, the arc of water below the point of reception, will be $\frac{1}{2}b\left(\frac{c^2 - 2x^2}{c - x}\right)$, and this is to be the greatest possible, or $\frac{c^2 - 2x^2}{c - x}$ = a maximum. By the principle of maxima and minima, this takes place when $x = c\,(1 - \sqrt{\tfrac{1}{2}})$ or $x = 0.2929\,c$. Accordingly, the arc $c - x$ must be a quadrant, and the arc $x = 37.27°$.

From this it appears that the wheel will produce its greatest effect when the diameter is so proportioned to the height of the fall that the water flows upon the circumference at a point 90° — 37.27° = 52.73° (nearly $52\frac{3}{4}$ degrees) distant from the summit of the wheel.

If then R be the radius of the wheel to the extreme edge of the bucket, and h the height of fall measured to the point where it may be delivered upon the wheel, and which may be called the effective height, then we shall have

$$\mathrm{R} = \frac{h}{1 + \sin 37\frac{1}{2}} = \frac{h}{1.605},$$

since $\sin 37\frac{1}{2}$ degrees $= .605$. We have also by reduction $\mathrm{R} = .623\,h$.

The effective height of the fall is less than the entire height H by as much as is necessary to give the water the required velocity, which may be taken generally at 10 feet in a second, or 1$\frac{1}{2}$ feet of fall.

The French mechanicians calculate a somewhat greater diameter for their wheels than that given by the foregoing rule. Instead of laying on the water at 52$\frac{3}{4}$ degrees from the summit, as is commonly done in this country, they lay it on at a distance of 60°, that is, 30 degrees above the horizontal plane passing through the axis of the wheel. Accordingly, $R = \frac{h}{1.5} = \frac{2}{3}h$. They also allow, as above recommended, 1$\frac{1}{2}$ feet of the fall to give the required velocity to the water.

No line of demarcation has yet been determined to separate this species of wheel from the breast wheel, except that this name is applied when the water is received upon the wheel at a greater distance from the summit than 52$\frac{3}{4}$ degrees. But it has not been decided when this rule ought to be set aside, and the wheel become a breast-wheel. A notion, not without foundation, prevails among millwrights that a wheel of large diameter is more advantageous than one of small diameter; in a wheel of large diameter the influence of the centrifugal force is less, and the mass in motion being greater, the movement is more uniform and may be proportionally slower, which, in the case of a low fall, is no inconsiderable advantage. There is, however, the disadvantage of additional friction upon the journals, and which, as these wheels are usually very broad, goes far to counterbalance the loss arising from centrifugal force.

As the question is therefore entirely one of practice, and incapable, we believe, of a theoretical solution, it may be stated as an opinion founded on a good example, that the diameter of the wheel, even for very large quantities of water, may be made to conform to the rule above given, down to 12 feet diameter. The example which we have in view, is a double wheel of that size, using at least 3000 cubic feet of water per minute very satisfactorily. The same size of wheel might be used till the fall descends to about 5 feet, when a wheel on the undershot principle will be found less expensive and equally efficient. Under these circumstances the wheel will act partly by the impulse and partly by the gravity of the water—that is, partly as an overshot, and partly as an undershot, wheel; its effect may therefore be ascertained by computing the effect due to the difference of level between the surface of the water at the penstock and the point where it strikes the wheel, and adding the result to the effect, realizable from the height of a fall equal to the difference of level between the point where the water meets the circumference of the wheel and the level of the tail-water, and which may be calculated by the methods above indicated.

SECTION III.—UNDERSHOT WHEELS.

By undershot we understand here those varieties of wheels which move chiefly by the direct impulse of the fluid. In construction they differ little from the bucketted wheel, except that the buckets are replaced usually by radial floats upon which the impulse of the current is received. They are, also, usually confined in an arc, below the level of the water-line, to confine and economize the motive power; but, as this arrangement is also common to bucketted wheels, especially when the fall is low, it cannot be regarded as a peculiarity. In this form of wheel, especially if the volume of water be considerable, the spider construction is, however, only admissible when the power is taken off at the circumference by a pinion placed slightly above the point of impulse and on the same side. There is, then, only the small portion of the sole-frame put on strain by tension, between the two points. But, when the power is taken off at the axis, the construction ought to be of the more rigid kind, otherwise, the continually changing direction of the strain, acting through a leverage equal to the radius of the wheel, will speedily prove fatal to the points of connection, if in any degree elastic.

The water is admitted upon the wheel by a sluice or shuttle in its immediate vicinity, as in the case of the bucket-wheel. What has been stated in reference to the loss of head experienced by the water passing through an opening in its course is therefore applicable in this case as before. It is also of moment, both on theoretical and practical grounds, that the sluice be placed as closely upon the wheel as other considerations will permit; and that the retaining cheeks of the aperture, inside of the sluice, be slightly contracted, answering to the natural contraction of the stream after passing through the orifice, in consequence of the resistance which it there encounters. The sides of the course or arc in which the wheel moves, must necessarily be parallel; but, immediately beyond the vertical plane passing through the axis of the wheel, the floor ought to deepen and the sides expand and leave the water as much space to diffuse itself over as possible. This arrangement is shown in Plate XCIV. as far as it is applicable with a sluice-framing entirely constructed of wood; but, when the construction is of iron, the confinement of the water may be made much more complete.

Supposing the floats to be placed radially, their breadth or depth in the direction of the radius, ought obviously to be such that in the rising of the water against the float which it first strikes, the portion which tends to pass over its superior edge shall not be thrown against the back of the succeeding float. Any action of this kind would manifestly be attended with a corresponding diminution of the effect of the wheel; and ought, therefore, to be avoided as, perhaps, the most serious error which is liable to be committed in this form of wheel. This source of loss may, however, be in general entirely avoided, by giving to the floats a depth of about three times the thickness of the lamina of water acting upon them. The thickness of the lamina is usually from four to six inches, giving the range of depth of the floats from twelve to eighteen inches. The velocity with which the fluid precipitates itself upon the floats, ought also to be taken into account in providing for its expansive movement. The distance of the one float from the other, measured upon the exterior circumference of the wheel, may be generally taken equal to the depth. Their number will, of course, depend upon the diameter of the wheel, and this is almost arbitrary. We will, however, endeavour briefly to indicate the general principle, which ought to be kept in view in fixing the diameter, without entering upon any strict investigation of the question.

As a consequence of the general theory already explained, it follows that the dynamical effect of the wheel is dependent upon the relation which the velocity of the floats bears to that of the water; but this relation is manifestly independent of the diameter. The velocity due to the current of water to be used is always an ascertainable quantity, and may therefore be regarded as known. Another determinate element of the calculation is the number of revolutions which it is desirable the wheel should make in a certain unit of time, as a minute, in order that the effect may be transmitted to the working points, with a rate of velocity the most advantageous for the particular purposes intended, and obtained in the simplest manner possible, and with the least quantity of intermediate geering. It is also important that the wheel have a velocity and dimensions rendering it capable of fulfilling the purpose of a fly at the rate of motion which it is intended to maintain. If we put u to denote the velocity of the extremities of the floats, and N the number of turns desired in a minute, the diameter will be expressed by

$$\frac{60\,u}{\pi\,N} = 19.1\,\frac{u}{N}.$$

And to obtain an effect approaching the maximum, we may assume $u = 2.4\sqrt{H}$; and therefore the diameter expressed in terms of the velocity and height of fall will be

$$19.1 \times \frac{2.4\sqrt{H}}{N} = \frac{46}{N}\sqrt{H} \text{ very nearly.}$$

Thus supposing the fall to be 4 feet, and the number of turns per minute required to be $8 = N$; then the diameter will be $\frac{46}{8} \times \sqrt{4} = 5\frac{1}{4} \times 2 = 11\frac{1}{2}$ feet nearly. This is nearly the minimum diameter of wheel which would under any circumstances be employed; 12 feet to 25 feet may indeed be taken as the usual range; but unless the volume of water be extraordinarily great—and then breadth is better than diameter—we cannot conceive of any advantage, other than may arise from some peculiarity in the nature of the machinery to be impelled, which may not be obtained with a diameter of 12 feet to 16 feet. It is, however, to be observed that the smaller the diameter the greater is the nicety of adjustment required to make the water yield its effect upon the buckets; and possibly the errors which have been committed in this particular have led to the common opinion that a wheel of large diameter is always in practice the most effective.

Before the introduction of Poncelet's system of curved floats, various attempts were made to increase the efficiency of the undershot wheel by placing the floats at some determinate angle with the circumference. And where the wheel moved in backwater, and especially in an unconfined channel, a certain beneficial effect was experienced. But in the ordinary cases of a confined course or arc with proper provision for tail-water clearance, these schemes do not seem to have been attended with any advantage. According to Bossut's experiments, indeed, the result appears to indicate that the radial float is the most efficient. Taking the effect obtained with the radial float as unity, the results which he obtained with the angular positions noted, are thus stated:

Angle of float,	0°	8°	12°	16°
Comparative effect,	1	0.949	0.956	0.998

Showing that at least no advantage is derived from *feathering the floats*, as it is denominated. In an indefinite volume of fluid, the case is, however, different; the inclination of the floats favours, not the action of the impelling fluid, but their disengagement. This is manifest when it is observed that as soon as the radial float passes into water, having a less velocity than its own, it begins to be retarded; and besides, in rising to the surface, it tends to lift with it a portion of the fluid, which, acting by its weight in the contrary direction to that of the wheel, also proportionally diminishes the useful effect.

This is sometimes obviated at great expense, in large wheels placed in situations where the fall is very small and liable to be flooded, by rendering them capable of being raised and lowered at pleasure, in conformity with the state of the river. The mechanism for this purpose is generally worked by manual labour; but sometimes also it is rendered self-acting, as in the case of some wheels worked by the tide in situations where the tidal oscillations are considerable. In these the floats are usually, though not always advantageously, inclined to the radius to assist the other arrangements.

Various other schemes for increasing the efficiency of impulsive wheels have been resorted to. One of these was to place a ledging on the ends of the floats to confine the action of the fluid; but with very little beneficial effect; and, it is obvious, that if any arrangement of this kind was likely to be of advantage, it could be most effectually secured by placing the floats within shrouds, as in the common bucket-wheel—a form of construction not uncommon. Another supposed improvement, still common in some parts of the continent, though never introduced into this country, and only applicable to very narrow wheels, is to form the floats of cylindrical arcs with the axis of the cylinder in the direction of the radius of the wheel, and the concave face of the arc opposed to the motion of the water, thus ⟶). This arrangement is stated to possess certain advantages; but we cannot conceive that they can possibly be so marked as to compensate the additional cost of construction; and we must still believe that the plain float with shrouding is both the simplest and most complete of all the deviations which have been attempted, excepting, indeed, M. Poncelet's application of the curved float, and even in this the advantage does not so much consist in the form of the float, as in the other beneficial adaptations with which it is associated.

To understand the action of the water upon the floats of a wheel of the kind under discussion, it is necessary to observe that the moment the sluice is raised, the fluid precipitates itself against the first float which obstructs its passage, and, in consequence, an accumulation takes place, which ultimately succeeds in putting the wheel in motion, and gradually accelerates its velocity until an equilibrium is established between the force of the current and the resistances to be overcome. In proportion as the velocity of the wheel increases, the pressure of the current becomes less, since this action is proportional to the relative velocities; and very soon the acceleration, which gradually diminishes, becomes imperceptible, and finally ceases; and the wheel after a certain number of revolutions, in consequence of the velocity impressed upon it, and in consequence also of its inertia, continues to revolve as of itself, either with a motion perfectly uniform, or with a velocity oscillating between limits imposed by the varying nature of the resistances, and which may be reduced in effect to a mean continuous velocity always ascertainable.

Supposing the wheel to derive its effect entirely from the impulse of the current, and to obtain no advantage from confinement in an arc, by which a certain amount of the weight of the fluid is made to act in aid of the impulsive force, the dynamical effect, considered theoretically, of any given weight W of water upon a float receding before the stream with a velocity of v feet in a second, is expressed by

$$\frac{W}{g}(V - v)\,v.$$

But it may be questioned whether the same effect will be produced upon a suite of floats presenting themselves successively to the current, usually two and three at a time, and under various angles of inclination. On this point we derive our most important information from experience; but admitting in the meantime that the action of the impulse, although not equal, is of the same kind, and susceptible of an expression of the same form as that above, we shall ultimately succeed in comparing the results of experiment with those of calculation.

In the expression above, when the wheel is moved by a confined current, v is the only variable quantity. If $v = 0$, the effect is also cipher, for a machine which does not move yields no power. The power will also be cipher, when $V = v$ since, as before remarked, if the wheel move at the same rate as the current, it can receive no motive effect from the fluid. It is still more obvious that v cannot be greater than V. The limit is therefore between $v = 0$ and $v = V$, and between these extremes there will be a *maximum* of effect. If, then, we differentiate the variable part $(V - v)\,v$, and equate to zero in the usual manner, we have

$$V\,d\,v - 2\,v\,d\,v = 0; \text{ whence } v = \tfrac{1}{2}V,$$

showing that the effect of the wheel is the greatest possible when it moves with half the velocity of the stream.

Now, the pressure of the water upon the floats being $\frac{W}{g}(V - v)$, this will also be a measure of the resistances (including all the passive resistances) overcome by the wheel, since the moments of pressure must obviously be equal in every case of equilibrium; hence, if in this expression we substitute for v its equivalent $\frac{1}{2}$ V, we have, as the measure of the load when the dynamical effect is a maximum,

$$\frac{W \times V}{2\,g}.$$

And the dynamical effect being equal to the load multiplied into the velocity of motion, viz., $v = \frac{1}{2}V$, we have, using this last as the measure of the velocity of the wheel,

$$\tfrac{1}{2}\,W \times \frac{V^2}{2\,g} \text{ equivalent to } \tfrac{1}{2}\,W\,H,$$

when H is put to denote the height of fall due to the velocity V, as before explained (p. 207).

If, therefore, V be the whole velocity of the stream, and H the entire fall due to that velocity, this result shows that the greatest possible effect which can be realized from a wheel moved entirely by the impulse of the fluid, is only half of the mechanical power of the water expended; that is, considering both cases theoretically, is only equal to half the effect which a wheel acting entirely by the gravity of the fluid, ought to realize. But even this moiety is subject to reduction, and can be only distantly approached in practice.

We do not, unfortunately, possess many experiments upon which we can implicitly rely, with wheels of this kind; we have many of a mixed kind, in which the effects of impulse and gravity are combined; but few in which the impulsive action alone appears. Those of Mr. Smeaton, indeed, stand nearly alone in importance and accuracy; and, fortunately, they are complete in the necessary data. Although the model apparatus with which they were made was small, the well-known accuracy of the experimenter and the purpose for which the investigation was undertaken, warrants the confidence which they have long commanded. The wheel was the same in diameter as his overshot model, viz., 2 feet; and was, indeed, adapted to the same apparatus. The power was measured directly by raising a weight vertically by a cord over a pulley; and, perhaps, the only objection which can be validly urged against the results, consists in his neglecting the additional friction thereby produced at the journals of the wheel. The data for this correction is, however, furnished, and may still be applied.

We subjoin the author's table of results, the columns of which are fully explained by the *headings* placed over them:

No.	Height of water in the cistern.	Turns of the wheel unloaded.	Virtual head deduced therefrom.	Turns at the maximum.	Load at the equilibrium.	Load at the maximum.	Water expended in a minute.	POWER.	EFFECT.	Ratio of the power and effect.	Ratio of the velocity of the water and wheel.	Ratio of the load at the equilibrium, to the load at the maximum.	Experiments.
	In.		In.		lb. oz.	lb. oz.							
1	33	88	15.85	30.	13 10	10 9	275.	4358	1411	10:3.24	10:3.4	10:7.75	At the 1st hole.
2	30	86	15.0	30.	12 10	9 6	264.7	3970	1266	10:3.2	10:3.5	10:7.4	
3	27	82	13.7	28.	11 2	8 6	243.	3329	1044	10:3.15	10:3.4	10:7.5	
4	24	78	12.3	27.7	9 10	7 5	235.	2890	901.4	10:3.12	10:3.55	10:7.53	
5	21	75	11.4	25.9	8 10	6 5	214.	2439	735.7	10:3.02	10:3.45	10:7.32	
6	18	70	9.95	23.5	6 10	5 5	199.	1970	561.8	10:2.85	10:3.36	10:8.02	
7	15	65	8.54	23.4	5 2	4 4	178.5	1524	442.5	10:2.9	10:3.6	10:8.3	
8	12	60	7.29	22.	3 10	3 5	161.	1173	328.	10:2.8	10:3.77	10:9.1	
9	9	52	5.47	19.	2 12	2 8	134.	733	213.7	10:2.9	10:3.65	10:9.1	
10	6	42	3.55	16.	1 12	1 10	114.	404.7	117	10:2.82	10:3.8	10:9.3	
11	24	84	14.2	30.75	13 10	10 14	342.	4890	1505	10:3.075	10:3.66	10:7.9	At the 2d.
12	21	81	13.5	29.	11 10	9 6	297.	4009	1223	10:3 01	10:3.62	10:8.05	
13	18	72	10.5	26.	9 10	8 7	285.	2993	975	10:3.25	10:3.6	10:8.75	
14	15	69	9.6	25.	7 10	6 14	277.	2659	774	10:2.92	10:3.62	10:9.	
15	12	63	8.0	25.	5 10	4 14	234.	1872	549	10:2.94	10:3.97	10:8.7	
16	9	56	6.37	23.	4 0	3 13	201.	1280	390	10:3.05	10:4.1	10:9.5	
17	6	46	4.25	21.	2 8	2 4	167.5	712	212	10:2.98	10:4.55	10:9.	
18	15	72	10.5	29.	11 10	9 6	357.	3748	1210	10:3.23	10:4.02	10:8.05	The 3d.
19	12	66	8.75	26.75	8 10	7 6	330.	2887	878	10:3.05	10:4.05	10:8.1	
20	9	58	6.8	24.5	5 8	5 0	255.	1734	541	10:3.01	10:4.22	10:9.1	
21	6	48	4.7	23.5	3 2	3 0	228.	1064	317	10:2.99	10:4.9	10:9.6	
22	12	68	9.3	27.	9 2	8 6	359.	3338	1006	10:3.02	10:3.97	10:9.17	4th.
23	9	58	6.8	26.25	6 2	5 13	332.	2257	686	10:3.04	10:4.52	10:9.5	
24	6	48	4.7	24.5	3 12	3 8	262.	1231	385	10:3.13	10:5.1	10:9.35	
25	9	60	7.29	27.3	6 12	6 6	355.	2588	785	10:3.03	10:4.55	10:9.45	5th.
26	8	50	5.03	24.6	4 6	4 1	307.	1544	450	10·2.92	10:4.9	10:9.3	
27	6	50	5.03	26.	4 15	4 9	360.	1811	534	10:2.95	10:5.2	10:9.25	6th.
1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.	13.	

but it may be observed, that several other columns of ratios might be deduced from the data therein furnished, and which would still further illustrate the action of this order of wheels. The last column has reference to the aperture at the sluice for the admission of the water to the wheel. The holes in the scale were placed diagonally, and to these a pin was fitted, so that when the pin was in the same hole, the opening for the water continued the same for all the experiments of that series.

From this table we find, on comparing the effect $p\ v$ produced at the maximum with the product $\frac{1}{2}$ W h, in which h is the virtual or effective head, that the coefficient of reduction m is very nearly 0.64; consequently, we shall have

$$\text{E or } p\ v = 0.64 \times \tfrac{1}{2}\ \text{W}\ h = 0.32\ \text{W}\ h.$$

The ratio of $p\ v$ to W h, we observe, varies from 0.28 to 0.32, giving a mean of 0.30; this led Mr. Smeaton to infer that $\frac{1}{3}$rd of the force produced on the floats by the current, may be realized in the larger wheels.

If we compare the effect realized with the entire power of the water expended, we find that the ratio increases from 0.16 in the first experiment, when the total head was 33 inches, to 0.25 in the last, when the entire head was reduced to 6 inches. From this it therefore appears that the greatest effect which can be obtained, from a given head of water acting impulsively, is between a sixth and a fourth of the entire motive force expended; and in the case of large wheels, it is very doubtful whether even this last result can be obtained, although, as we have already seen, theory indicates as much as $\frac{1}{2}$ W H or double.

The ratio of the velocity of the wheel to that of the current gradually augments from 0.34 to 0.52, giving a mean of 0.43. Mr. Smeaton, however, takes 0.40 as the mean; and it is worthy of remark that Bossut, in an analogous series of experiments, also adopted the same number. It, however, seems to us, from the nature of the case, that the proper velocity will approximate much more closely to the maximum limit, and will not deviate greatly from 0.50 of the mean velocity of the current as indicated by theory. 0.45 will at least be in general cases a safe number to adopt in practice; that is, $v = 0.45$ V.

Another result worthy of notice is the weight or "load at the equilibrium," that is, the weight which is just sufficient to keep the wheel at rest against the force of the current. This, at an average, is little more than two-tenths greater than the load which the wheel was capable of carrying when yielding its maximum effect. According to theory it ought to be double, for, as already shown, the weight corresponding to $v = 0$ is $\frac{\text{W} \times \text{V}}{g}$, and that at the maximum is $\frac{\text{W} \times \text{V}}{2\,g}$.

The cases, to which these observations apply, are those in which the velocity of the wheel is adapted strictly to that of the current. But this is not always obtained, and accordingly the coefficient m being a function of v, fluctuates between extremes which it is impossible to comprehend in a general formula. However, when the velocity of the wheel does not fall below certain limits—from a third to two-thirds of that of the current—we may, without much chance of error, especially in excess, assume 0.60 as the coefficient, and accordingly we shall have as a general rule,

$$\text{E}=0.60\ \frac{\text{W}}{g}\,(\text{V}-v)\,v = \frac{1}{53.7}\ \text{W}\,(\text{V}-v)\,v = 1.16\ \text{Q}\,(\text{V}-v)\,v.$$

The velocity V with which the water arrives upon the floats cannot always be easily assigned. It experiences certain losses between the sluice and the point of impulse, but it is not perhaps possible to give a general expression of their amount, even for an individual case, and much less for different forms and conditions of construction. Independently of any reducing influence, we have $\text{V} = \sqrt{2\,g\,h}$, in which h denotes the difference of level between the surface of the water in the lead and the centre of percussion of the floats, and which can readily be measured. But from Mr. Smeaton's observations on this point, it appears that the loss is sometimes as much as a fifth of this velocity; and he further remarked that the difference between the actual and calculated velocities diminished as the vertical opening of the sluice was augmented. In some instances, indeed, where the volume of water was very great and the head small, he found that V hardly differed from $\sqrt{2\,g\,h}$. M. Poncelet also remarked the same circumstance—that the loss of velocity diminishes with the magnitude of the aperture through which the fluid issues. Even in the case of an opening of about $8\frac{1}{2}$ inches in height, and with a head of $4\frac{1}{2}$ feet, he had $\text{V} = 0.99\sqrt{2\,g\,h}$. This, however, supposes the sluice to be constructed to the best advantage; and to make a slight allowance for untoward circumstances, we may take $\text{V} = 0.95\sqrt{2\,g\,h} = 7.6\sqrt{h}$. Substituting this value of V in the preceding result, we have

$$\text{E} = 1.16\ \text{Q}\ (7.6\sqrt{h} - v)\ v.$$

And, again, putting for v its equivalent $\frac{1}{2}\text{V} = 3.8\sqrt{h}$, this expression is reduced to the following convenient form,

$$\text{E} = 16\tfrac{3}{4}\ \text{Q}\ h, \text{ or } \frac{1}{1971}\,\text{Q}\ h,$$

when expressed in units of horse power.

This is the case of the purely impulsive wheel; but in practice it is very rarely found that some slight amount of head cannot be reserved to act by its gravity. Under these circumstances an arc is formed concentrically with the wheel between the point at which the water strikes the wheel, and the lowest level terminating in the vertical plane passing through the axis. The clearance need not exceed .4 inch, and may, if the arc be carefully constructed, be reduced to .3 inch. To indicate the effect, let H denote, as before, the whole height of fall; and let h be that portion of it employed in generating the required velocity. After the water has struck the first float at the level $\text{H} - h$, it will descend by its weight through that height, producing an effect expressed by $\text{W}\ (\text{H} - h)$; and the effect of the impulse being as before $\frac{\text{W}}{g}\ (\text{W} - v)\ v$, we shall have as the sum of these partial actions,

$$\text{W}\ \{(\text{H} - h) + \frac{1}{g}(\text{V} - v)\ v\}.$$

But we formerly found in discussing the impulsive action in the case of the overshot wheel that

$$\frac{\text{W}}{g}\ (\text{V} - v)\ v = \text{W}\ (h - \mu\ h - h' - h''),$$

in which the three quantities μ, h', h'', have the significations then assigned. We may therefore, make use of this expression as more definite. It is, however, in this case subject to two corrections which may be thus exhibited.

In the first place, when the whole volume of water has expended its impulsive effect upon the first float which it encounters, it immediately begins to descend by its pressure to the bottom of the arc; but let us observe what takes place in the spaces between the floats during the descent. The arc sustains a certain amount of the pressure of the fluid, and there is moreover a certain amount of clearance between it and the radial extremities of the floats. A portion of the fluid will therefore escape through this space without producing any effect, since its pressure is entirely exercised upon the superficies of the arc. This must, consequently, be subtracted from W in the expression W (H — h). This loss cannot, indeed, be rigorously assigned, but may be pretty closely approximated by considering that the resistance experienced by this water against the face of the arc diminishes the velocity which gravity tends to give it, and that this diminution increases with its descent; also, that this velocity is further diminished by the continual entanglements, to which the water is subjected by the varying conditions of the intervals between the floats, and which likewise become greater towards the bottom of the arc; and, finally, that the velocity is altered by the continual mingling of the descending laminæ, corresponding to the several spaces between the floats and the varying positions of the portions of fluid therein contained. We may therefore conceive, with all these retarding influences in action, that the velocity of the ineffectual portion will not differ greatly from that of the floats; accordingly, in this state of things, if we denote by A the cross section of the plate of water falling upon the wheel, and by a that corresponding to the intervals between the extremities of the floats and the arc, then will $W \frac{a}{A}$ be the portion of fluid lost as regards the effect of pressure; hence, by subtracting this from W, the expression of the effect given above, we shall have

$$W \left(1 - \frac{a}{A}\right) (H - h).$$

In the second place, the portion of the base of the wheel which dips in the water contained in the lower part of the course, loses there a part of its weight equal to the weight of water displaced. In consequence of this loss the equal distribution of the weight of the wheel about the axis of rotation no longer exists; and the wheel tends to turn in a direction contrary to that of the current. If we represent by p' the diminishing influence of this tendency, this will be a new resistance which the wheel has to overcome, and which ought, consequently, to be added to these other resistances of which the sum is p. We shall then have, taking m as the coefficient of reduction of the results of calculation to those of observation,

$$(p + p')\, v = m\, W \left\{(H - h) \left(1 - \frac{a}{A}\right) + h + \mu h - h' - h''\right\}.$$

In practice this formula may be considerably simplified. The quantities p' and $1 - \frac{a}{A}$, supposing the construction judiciously and carefully finished, will be very nearly proportional to the power of the wheel, that is, to W; they may, consequently, be comprised in the value of m. We have also before shown that the quantity $\mu h + h' + h''$ is always greater than $\frac{1}{3} h$, and differs little in ordinary cases from $\frac{2}{3} h$. Hence our formula may be reduced by these substitutions to the convenient form,

$$E = m\, W \left(H - \tfrac{2}{3} h\right).$$

In this the indefinite quantity is m, and, perhaps, the best authenticated experiments, by which its value may be assigned for the particular case assumed, are those of M. Morin, on a wheel constructed by Messrs. Aitken and Steel for the crystal works of Baccarat, in the Department of Meurthe in France. The diameter of the wheel is 13 feet 3 inches; its width parallel to the axis 12 feet 9½ inches; the number of floats 32, of which the breadth in the direction of the radius is 1 foot 4 inches. The whole fall is 6 feet 9 inches, and the versed sine of the arc 6.04 feet. The water is thrown upon the wheel over the waste-board of a sluice, of the same width as the wheel. The results varied with the thickness of the lamina of water admitted upon the wheel, as exhibited in the following table.

Velocity of the wheel in feet per second. v.	Thickness of lamina of water upon the waste-board of the sluice.	Ratio	
		of effect to power expended W H $\frac{p\,v}{W\,H}$.	of effect to virtual head $H - \frac{2}{3} h$ $\frac{p\,v}{W\,(H - \frac{2}{3} h)}$
	Feet.		
7.65	0.719	0.707	0.762
3.83	0.711	0.734	0.792
3.18	0.711	0.726	0.783
2.71	0.714	0.720	0.777
2.40	0.714	0.716	0.773
2.13	0.718	0.700	0.755
	Means.	0.717	0.772

From this table then, it appears that $m = 0.772$; but as this is reputed to be a particularly well constructed wheel—considerably above the average—we may be generally safe in taking $m = 0.75$, by which our formula is reduced to

$$E = 0.75\, W \left(H - \tfrac{2}{3} h\right).$$

From the same table it appears that the ratio of the effect to the whole power expended, is 0.717; this is a good result, and warrants us in taking, as the general expression of effect for a wheel of ordinary character under like circumstances, 0.65 W H.

The effect of curving the floats, as in M. Poncelet's wheel, is thus indicated: Supposing, in the first place, that the wheel is at rest, and that a film of fluid arrives horizontally with a velocity V upon the lower edge of the float, in continuing to advance it rises along the curve, and during its elevation, the velocity which it possessed, is gradually diminished, and becomes nothing when it has attained a height expressed by $0.0155\, V^2$. The velocity is not, however, lost; it is simply changed into gravity, in obedience to which the fluid immediately begins to descend upon the curved surface of the float, over which it ascended, and quits it with the same velocity V, which it possessed at the moment it arrived upon it. This velocity is acquired by falling from the height $0.0155\, V^2$, and under the circumstances we have supposed to exist, its direction would be contrary to that first impressed upon the fluid. Let us now assume that the wheel turns with a velocity of v in a second at its periphery. When the filament of fluid, having the velocity V, shall have arrived at the float, it will then have a relative velocity of $V - v$, and it will only be with this velocity that it will commence to ascend upon the curved surface of the float; it will therefore rise to a height of only $0.0155\ (V - v)^2$, and after descending, will quit the lower edge of the float with the same velocity $V - v$. But this element will now have itself a velocity v in the contrary direction, for it partakes of the motion of the wheel; the absolute velocity with which it escapes will therefore be $V - (v + v)$. Consequently, if $v = \frac{1}{2} V$, the absolute velocity of escape will be $V - V = 0$, showing, that if the velocity of the wheel be half of that with which the water arrives, its absolute velocity in quitting the floats will be nothing. We have, therefore, the case of a motive current, which experiences neither shock nor loss of velocity at the moment of impulse upon the wheel, and which possesses none at the moment it quits the float; it has then expended all its movement upon the wheel, and communicated to it all its force. The two conditions, shown to be unattainable in the bucket, and common impulsive wheels, is therefore theoretically attained with this arrangement, so that, if W be the weight of water, and h the height of fall due to the velocity V, we shall have as the expression of effect $W\,h$.

But although this may be nearly true for a simple film, it is not true for a volume or sheet of water of a certain thickness. Those molecules which strike the floats, making an angle more or less great with the element struck, lose both a portion of their velocity and force; and at the moment when the mass of particles quit the float upon which they have expended their action, their direction is not exactly opposite. Besides, as with all wheels which revolve in an arc, a part of the motive fluid escapes without yielding any useful effect. We may, therefore, conclude that the real effect is not $W\,h$, but $m\ W\,h$, in which m is some fraction less than 1.

A series of experiments was undertaken by M. Poncelet for the purpose of determining this fraction, that is the ratio between the actual effect realized and the power expended. The following table contains the most important conclusions. The wheel, it may be observed, had a diameter of $11\frac{3}{4}$ feet; 30 floats of $12\frac{1}{2}$ inches depth in the direction of the radius, and 25 inches breadth between the shrouds.

Rise of sluice in inches.	Ratios.		
	$\frac{v}{V}$.	$\frac{p\,v}{W\,h}$.	$\frac{p\,v}{W\,H}$.
3.937	0.46	0.51	0.46
8.268	0.52	0.70	0.56
8.661	0.60	0.68	0.56
7.874	0.52	0.60	0.52
11.969	0.69	0.81	0.55
"	0.61	0.74	0.55
"	0.59	0.63	0.52

From these experiments and observations M. Poncelet concludes :—

1°. That the velocity of the wheel which gives the maximum of effect is 0.55 of the velocity of the current; but that it might be varied from 0.5 to 0.6 without any marked disadvantage.

2°. That the dynamical effect is not under $0.75\ W\,h$ for low falls with large volumes of water, and may be taken at $0.65\ W\,h$ when the volume of water is small and the fall considerable.

3°. That this same effect, compared with the entire force expended, namely, $W\,H$, may be taken at 0.60 in ordinary cases, and at 0.50 when the rise of the sluice is very small.

For those cases which ordinarily present themselves in practice, and, supposing the wheels constructed with due care, and to be adjusted to velocities differing little from .55 of the current, we may therefore take

$$E = 0.75\ W\,h \quad \text{or} \quad E = 0.60\ W\,H.$$

Comparing this result with that determined for impulsive wheels having radial floats, it appears that the effect is more than doubled. This conclusion, to which we arrive in both cases, by experimental guidance, ought of course to decide which of the two forms of wheel ought to be employed in general cases. It is admitted, that M. Poncelet's wheel involves a more precise acquaintance with the nature of the force employed than the common float-wheel; but nothing beyond the application of a few rules, which any millwright may readily comprehend and apply. These have in part been given in our description of Plates XCIV and XCV. In this last, the rule for describing the curvature of the floats is given in Fig. 2. The extreme and interior circles of the shrouds being drawn such that $o\,k = \frac{1}{4}$ the effective fall when not more than $4\frac{1}{2}$ feet, the circle $m\,n$ is described with a radius determined by the following considerations:—From the point k at which the water is supposed to meet the exterior circumference of the wheel, draw the line $k\,p$ perpendicular to the direction of the fluid. It will form an angle of 24° to 28° with the radius. In that line take a point p equal to about a sixth of its length between the circles of the shrouding, within the inner circle, and through that point from the centre of the wheel describe the circle $m\,n$. Then will $p\,k$ or $p\,q$ be the radius of the curved float $k\,q$; and similarly all the radii of the other floats will terminate in that circle. Having determined the number of floats, and marked their extremities upon the external circle of the wheel, draw radii from these points to the axial centre, and upon the circle $m\,n$ set off the corresponding distances from these radii equal to $l\,p$, and the points thus found will be the centres of curvature of the floats. The distance between the floats will be about half that recommended when placed radially, and ought to be formed of sheet-iron both for convenience of making and subsequent economy of action.

The mode of constructing the arc at the base of the wheel has been explained in describing the Plates referred to; it is further only necessary to observe that every care ought to be employed to absorb as little as possible of the velocity of the water previous to the moment of impulse, and to provide for its escape when it has expended its force upon the wheel.

It is also to be understood that this species of wheel, or, more correctly, the mode of supplying the water, will not be economical for falls of more than $4\frac{1}{2}$ feet; when the fall exceeds this limit, advantage ought to be taken of its weight as well as of its impulsive force. We conceive, however, that the form of wheel is itself well adapted to this double purpose; but the water, instead of issuing from the under edge of the sluice-plate, ought to be directed over it, as over a waste-board; and the height of the arc ought, at the same time, to be proportionally increased.

Wheels which move in an indefinite current of water, as a river, are usually of a heavier construction than those we have been considering; but differ only in that respect, and in the inclination of their floats, from the common impulsive wheel. It is usually found of advantage to make them of a diameter of 15 to 20 feet, with 12 to 16 floats, of which the best inclination appears from experiment to be 30°. Their best velocity—that at which the effect is a maximum—is a third of that of the current; and, under these conditions, it will be found that they yield an effect of about $.006 \text{ W V}^2$ of the water received upon the area of the floats—that is, about $\frac{2}{5} \text{ W } h$ if $h = 0.0155 \text{ V}^2$. This result may seem, at first sight, surprising, when it is remembered that the effect of the undershot wheel working in a confined rectilineal course, does not yield more than $\frac{1}{3} \text{ W } h$; but it is to be observed that in this last we include the whole volume of water acting; whereas, in the other, we take into account only the quantity received upon the floats, without reference to the large amount which escapes without producing any effect whatever, and which we cannot attempt to estimate

This species of wheel is of very rare occurrence; yet there are numerous situations where it might be employed with good effect.

SECTION IV.—HORIZONTAL WHEELS.

This name comprehends a large variety of wheels, exhibiting often more ingenuity in their design than judgment in the application of those forces upon which their action depends. Many of them are matters rather of curiosity than of utility; and it is further to be observed, that few of them have found their way to this country except in treatises on hydraulics of continental birth. The only two specimens to which we conceive it worth while to direct attention are, the Turbine of M. Fourneyron and the Reaction wheel of Mr. Whitelaw—not inappropriately called the Scotch Turbine. Both of these are wheels belonging to the first class, from the percentage of effect which they are shown to realize; and, brought into competition with other varieties, in which the same principles are attempted to be carried out, must speedily replace them. We give precedency to the Turbine, not that we conceive it to be in any respect superior; but because its mode of action is more nearly allied to wheels of the kind we have been considering, than the Reaction wheel, which introduces a principle only briefly discussed in our introductory section.

Fourneyron's Turbine.—PLATES LXXXIV.—LXXXVII.

In 1834 M. Fourneyron received the prize of 6000 francs offered by the Society for the Encouragement of the Arts at Paris, for the construction of the best horizontal wheel on the large scale. This was his first Turbine erected at Pont, on the Oguon. In consequence of this decision the Turbine excited great attention and discussion among the continental *savants;* and it must be remarked, that matters of practical utility are more subjects of interest among scientific men, especially in France, than in this country, where every thing of a technical character is considered to belong exclusively to the workshop.

In an elaborate report on this machine submitted to the Academy of Sciences of Paris by M. Poncelet, it is stated that the essential quality of the Turbine consists in its high velocity, and its capability of working under water without much loss of effect. The expedient of bringing the water horizontally over all the interior circumference of the wheel, and of making it issue through the greater exterior circumference, allows also a large expenditure of power with a machine of very moderate diameter. Finally, it operates favourably under almost any fall and at any velocity, without suffering any reduction of its effect from the hydrostatic pressure of the water—which is stated to be a source of great inconvenience in wheels of this class.

After remarking on the nature of the investigation, the report proceeds to state that, although the Turbine is not generally susceptible of producing what is called the maximum of absolute effect, yet it affords results which come very near to it; that the relation of the useful effect to the power expended, and also the velocity of the wheel to that due to the effective fall, are wholly independent of the height of this fall, and of the depth to which the revolving part may be immersed in the tail-water; and that the values of the ratios of the effects vary but little for angular velocities sensibly deviating on both sides of that at which the relative effect is a maximum. These conclusions, it is stated, fully agree with the results of known experiments—and, it may be here observed, that the calculations are compared chiefly with the experiments made by M. Morin on a Turbine erected at Mülbach by M. Fourneyron. In this comparison it is found that the values of the ratios assigned by calculation are far from decreasing for high velocities, with that rapidity which they are shown to do in practice. This, as also the circumstance that the diminution of useful effect in reference to very small apertures, is likewise less sensible in the results obtained by calculation, affords proof that the immersion of the wheel in tail-water, and other concomitant conditions, must be included among the elements of the problem before a complete solution can be obtained. Besides, the same agreement is perceivable in the comparison of the theoretical and actual expenditure of fluid, when we take into account the disturbing influence which may be attributed to those circumstances.

M. Poncelet states that he has endeavoured approximately to include, not only the resistance encountered by the Turbine while moving in tail-water, but also the influence which may be ascribed to the annular jet, or the vacuum produced between the reservoir and the upper crown of the wheel, as also that due to the intermediate diaphragms sometimes introduced by M. Fourneyron. It may, indeed, be conceived that when the latter is put in motion, the internal pressure will be greater or less than the circumambient fluid, for when the motion is rapid there will be a certain amount of aspiration or *suction* produced by the revolving part; and, on the contrary, when the velocity is below a certain rate, the water of the internal crown will press upon and tend to expel that in the turbine, thus, in both cases, modifying the dynamical effect of the machine and the quantity of water discharged by it. This will, of course, be more sensibly felt when the velocity of the annular jet is increased at the apertures of the fixed crown by lowering the sluice-plates, and when the velocity of the wheel approaches its extreme limits.

On the other hand, when intermediate diaphragms are introduced, it is found that with small sluice openings, the fluid received into the upper divisions, from the water in which it is immersed, is subjected to the action of centrifugal force, and, consequently, tends to escape with a velocity increasing as that of the wheel, thereby producing a constant afflux of the waste-water from the exterior through the empty courses of the revolving part, an effect which does not take place in the lower division of the turbine, because the feed-water flows into it uniformly and with a constant pressure greater than that without.

These details show that the theory and *modus operandi* of turbines are delicate questions in themselves; the analytic investigation of the different conditions implied, lead to equations highly complex; and, indeed, each case would require a particular calculation for itself upon the method of successive approximations. In practice, the effect is liable to an indefinite diminution from an improper arrangement of the parts, and especially by a false estimate of the velocity, of the quantity of water which it is capable of discharging, or of the areas of the inlet apertures suited to the outlet orifices in the revolving part of the machine. M. Poncelet further adds, that the excellent results obtained by M. Fourneyron are due to his fully understanding the real constitution of the machine, and his long experience in the practical application of his theoretical knowledge.

The peculiar character of the machine is sufficiently explained in the description of the plates referred to, and which are supposed to represent one of the inventor's most successful applications of his principle; but, in order to bring the value and relation of the forces more closely into view, the action of the water may be here briefly indicated. Supposing the annular sluice to be so far let down as entirely to close the spaces *d*, which form the communication between the interior cistern *b* and the channels *c* of the revolving disc *e*, which is the *turbine*, properly so called, if the sluice between the reservoir and the supply pipe *a* be opened, the water will precipitate itself into the cistern *b* and entirely fill it. The pressure on the interior of this cistern, as well as on the annular sluice at the orifices *d*, will be in proportion to the depth from the higher level of the water, and, therefore, for a unit of area of the surface acted upon, the pressure will be directly as the height H. If, then, the sluice be raised, the water rushes into the channels *c* with the velocity due to the head of pressure, and in the direction prescribed by the guide curves, and impinging against the diaphragms of the channels *c* causes the disc *e* to revolve in a direction opposite to the direction of impulse, and finally escapes by the external extremities of the channels at the greater circumference of the turbine-disc.

The lower divisions of the sluice-ring, it may be remarked, are considerably increased in thickness, and rounded to avoid the contraction of the veins of fluid issuing into the channels *c*, and which would take place, if no provision were made for correcting the oblique motions impressed, when the water is projected through the apertures at the extremities of the guide-curves in a horizontal direction. If the plates above these orifices were thin, and such as would be made, in the common routine of construction, the areas of the jets of fluid, as we have on a former occasion shown, would converge, and thus enter the channels *c* under angles, of which the values can be determined by theory, and which, it is known, detract from the dynamical force of the jets. This effect cannot be entirely avoided, and accordingly the velocity with which the water enters the channels is not $\sqrt{2\,g\,H}$, which theory indicates, but is given by the equation

$$V = \frac{\sqrt{2\,g\,H}}{\sqrt{1 + \left(\frac{1}{m} - 1\right)^2}}.$$

And as m may be taken generally without sensible error equal to $\frac{3}{5}$, this rule may be reduced to the simple form $V = 6.66\sqrt{H}$. Now if we put U to denote the velocity with which the water escapes from the external orifices of the turbine; $A_{,}$ the sum of the areas of the jets at the sluice-plates, and $A_{,,}$ the sum of the areas of the external jets, then we have $Q = A_{,}V$, and $V = \frac{A_{,,}}{A_{,}}\,u_{,}$ when $u_{,}$ is put for the relative velocity of the water to that of the turbine at the exterior circumference. Hence, if v be the velocity of the inner circumference of the turbine, of which the radius is r, and θ the angle which the water makes with that circumference on entering, the relative velocity u with which the fluid enters, will be expressed by

$$u = \sqrt{\left\{\frac{A_{,,}^2}{A^2}\,u_{,} + v_{,}\left(v_{,} - 2\,\frac{A_{,,}}{A_{,}}\,u_{,}\right)\cos\theta\right\}},$$

since that velocity is the resultant of the two velocities V and $(-\,v_{,})$; and the absolute velocity of escape being the resultant of the relative velocity $u_{,}$ and of $(-\,v)$ will be

$$V = \sqrt{u_{,}^2 + v^2 - 2\,u_{,}\,v\cos\phi},$$

ϕ being the angle at which the water escapes at the external circumference. If then, R be the external radius, and $b = \frac{R}{r}\sin\phi$ and $c = \frac{A_{,,}}{A_{,}}\sin\theta$, we shall have as the sum of all the losses of *vis viva* in the machine,

$$\frac{W}{g}\left(u^2 + b^2\,u_{,}^2 - 2\,b\,c\,u_{,}^2\right);$$

and equating the *vis viva* of the water at its escape to that possessed by it on entering, augmented by double the quantities of action impressed and diminished by the amount lost in passing through the orifices and channels, we have

$$u_{,}\sqrt{1 + \gamma} = \sqrt{2\,g\,H + \omega^2\left(R^2 - r^2\right)},$$

in which $\gamma = \frac{A_{,,}^2}{A_{,}^2} + b^2 - 2\,b\,c,$

and, therefore, $Q = \frac{A_{,}}{\sqrt{1 + \gamma}}\sqrt{2\,g\,H + \omega^2\left(R^2 - r^2\right)}.$

ω being the angular velocity of the revolving disc. The first of these factors for the quantity of water discharged thus depends upon the dimensions of the machine; the other takes for its first term, the action of gravity to produce the velocity with which the water escapes from the cylinder, and for the second, it takes the effect of the centrifugal force. This equation shows, moreover, that, in consequence of this last force, the expenditure of water exceeds that which is due to the simple difference of height between the higher and lower levels, denoted by H, and that this excess will continually increase with every increase of the angular velocity ω of the machine.

The expression of effect deducible from these elements is complex and of little practical moment, as it differs very widely from that deduced by experiment. According to the investigations of M. Morin, the maximum coefficient seems to range from .60 to .80. We, however, believe it to be quite improbable that a machine, subject to so many deteriorating influences as the turbine, can possibly yield a result of .80 WH; and we believe that M. Morin has, since these experiments were made, revised the formula employed to determine the quantity of water used, and increased his coefficient in some cases fully ten per cent., thereby deducting an equal amount from the coefficients of effect of the machines for which the formula was employed. The coefficient given by M. Fourneyron himself, by M. Morin, and others, who have investigated the merits of this species of machine, is .70, making $p\ v = .70 \mathrm{WH}$. The effect of that at St. Blazien, depicted in the plates before referred to, is said to exceed .75 W H. This, from its great power (40 horses) and small size, is, perhaps, the most extraordinary hydraulic machine ever constructed, and has conferred much deserved celebrity on the inventor of the turbine.

The construction of the machine depends upon the application of a few fundamental principles. Like all other hydraulic motors, its size ought to be proportioned to the effect which it is intended to produce—that is, in effect to the quantities W or Q and H. Thus the interior diameter D, one of the principal dimensions, is directly as the ratio of these two quantities; and as the turbine ought to be capable of expending the volume of water Q, arriving to it with the velocity V, the orifices must have an area, determined from the condition $Q = A\ V$, in which we denote by A the sum of the orifices of admission. On the water arriving in the same time upon all the whole interior circumference of the turbine, A will be equal to that surface (after subtracting the area occupied by the thicknesses of the diaphragms), and, consequently, will be equal to $\pi\ D\ d$, in which d denotes the depth of the courses. The proportion fixed by M. Fourneyron, is $d = \frac{1}{7}\ D$; and, therefore, by making this substitution, we shall have

$$A = \tfrac{1}{7}\,\pi\, D^2 = 0.45\ D^2.$$

But $Q = A\ V \qquad = 0.45\ D^2\ V.$

And $V = 6.66\sqrt{H}$; therefore $Q = 3\ D^2\sqrt{H}.$

From this we have the diameter $D = \sqrt{\dfrac{Q}{3\sqrt{H}}}.$

This value of D ought, according to the views of the inventor, to be further affected by a coefficient, to allow for the entanglements which the fluid experiences in the cylinder and in entering the turbine, and for the effect of the obliquity with which the water is thrown by the diaphragms upon the moving circumference. This coefficient being introduced according to the practice of M. Fourneyron, we have

$$D = 1.3\sqrt{\frac{Q}{\sqrt{H}}}.$$

It is here assumed that Q is the greatest volume of water which the machine is capable of discharging; but it is to be understood, that smaller quantities may be employed, and that the machine is capable of working with almost any less quantity without losing any remarkable amount of its proportional efficiency.

The diameter D may thus be taken as a function of the power of the machine, that is of E the dynamical effect in units of horse power. Now, assuming the machine to realize 75 per cent. of the power which it expends, and that Q is the volume of water supplied in 1 minute, we have

$$E = \frac{Q\ H}{700}. \qquad \text{Hence, } Q = \frac{700\ E}{H}.$$

And, substituting this value of Q in the expression for D above given, it becomes

$$D = 1.3\sqrt{\frac{700\ E}{H\sqrt{H}}} = 35\sqrt{\frac{E}{H\sqrt{H}}}.$$

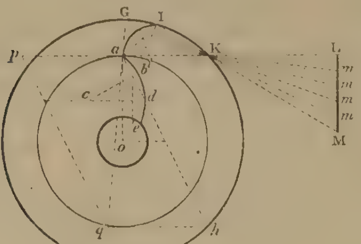
The exterior diameter, in the practice of M. Fourneyron, varies from 1.2 D to 1.44 D. For turbines of large diameter, 6 feet and upwards, the first of these coefficients is taken, and for smaller diameters, the last.

The number of channels, of course, also varies as the diameter, but not proportionally. In the rules published by the inventor 36 is given as a constant number with the same number of guide-curves on the interior disc; but in some of the machines of later construction these numbers are reduced. D'Aubuission mentions turbines which he has examined having as few as 18 channels with from 16 to 9 guide-curves. Jariez gives the following rule for the number:—Divide the interior circumference by the height d, and the quotient number which results is the number of channels in the turbine. If this number be comprised between 18 and 24, its half represents the number of fixed guide-curves; if it be greater than 24, then a third of it will be the number of these fixed compartments. It is, however, easy to perceive that this rule must only be a distant approximation; but even an approximation is better than no rule where theory seems insufficient to determine the question. The number, according to Prof. Rühlman, depends principally upon the available quantity of water; they must be greater as more water is discharged in a given time. In any case a large number of channels is an advantage, when they are formed of thin sheets, as thereby a greater number of filaments of water act directly upon their surfaces, and not indirectly through a mass of other water interposed.

We have above, following the rules laid down by M. Fourneyron, giving the depth of the channels as a seventh of the inside diameter of the turbine; but when the sluice is raised only a small part of this height, as it must be at times when the supply of water is scarce, the effect is not only absolutely less: it is relatively so on account of the water losing a portion of its force in diffusing itself over too much space. To avoid this, M. Fourneyron, in some of his last constructed machines, has divided the turbine, as before intimated, horizontally into two or three stages, by means of thin plates of sheet iron placed in the channels.

The curvature of the water channels of the turbine and of the guide-curves in the fixed crown, may be determined by the following mode. Describe the interior circumference with radius $o\,a = \mathrm{D}$, found as above directed; also the external or greatest circumference of the turbine with radius $o\,\mathrm{G} = 1.4\ \mathrm{D}$. These circumferences being described, draw $a\,h$ making $\sin h\,a\,o = \frac{\mathrm{V.}}{2\,v}$. From the centre o draw $o\,d$ making with $a\,o$ an angle $d\,o\,a = d\,a\,o$; and from the point e, where $o\,d$ cuts the circle representing the tube or pipe through which the spindle of the turbine ascends, take $e\,b$ parallel to $o\,a$; from b draw $b\,c$ perpendicular to $a\,d$, and from d draw $d\,c$ perpendicular to $e\,b$; the point c where these two perpendiculars meet, will be the centre of the fixed or guide-curve $e\,d\,b\,a$.

Fig. 11.



To find the curve of the vertical diaphragm a I of the turbine, draw $a\,p$ a tangent to the point a; and let $a\,p$ and $a\,h$ be proportional to the velocities v and V of the turbine and water: the diagonal $a\,q$ of the parallelogram constructed upon these two lines will be the direction of the first element of the curve. Prolong $a\,q$ to G, making a G perpendicular to a L, which is $p\,a$ prolonged indefinitely, cutting in K the exterior circumference of the turbine-disc. The point I at which the extremity of the curve terminates, is $\frac{2}{5}$ of G K. The two extreme points being thus found, the curvature is determined as follows. From the point K as a centre, and with a radius I K, describe the arc of a circle I i, and prolong indefinitely the right line I K. Measure the line $a\,i$ with any scale of equal parts, and divide the number expressing its length by $1 - \cos$ M K L. and the quotient number of this division, taken in units of the same kind as $a\,i$, will express the length K M. From this point M draw M L perpendicular to K L; divide M L into any number of parts as m, m . . . . as many as convenient, and the more the better; from each of these points draw a straight line passing through the point K, and with the length I M in the compasses mark off equal distances from the points m . . . . on their prolongations beyond K, and the points thus marked off will be points in the curve a I, which may accordingly be traced through them.

If these rules be strictly followed a very good turbine will be produced, assuming the workmanship to be of proper quality; but, it is to be observed, that there are local circumstances and conditions often interfering, which require modifications to be introduced, for which no rules can be laid down. An accurate knowledge of hydrodynamical science, combined with experience, can alone warrant any departure from the precepts furnished by the inventor, in whose hands alone the machine has given promise of high capability. He has published very little on the subject beyond his first Memoir to the Society, and, indeed, retains the knowledge of the constructive details, in the meantime, as far as possible to himself, probably with the intention, at some future time, when his experience is matured, of giving a more precise account of his rules and practice than can be expected from the brief period during which the invention has been in operation. It is, however, to be observed, that as M. Fourneyron makes the construction of turbines a business, it is scarcely to be expected that he will be anxious to transfer the results of his study and experience into other hands, so long as he can successfully maintain a monopoly of his invention. Many attempts have been made on the continent to rival his mode of construction, but, generally, they have led only to disappointment; and, we are by no means sanguine that any efforts made in this country to displace the common water-wheel by a machine requiring so much nicety of technical detail, and so much preliminary knowledge of hydraulic principles, will be attended with better success.

Whitelaw's Reaction Wheel:—Plates LXXXI.—LXXXIII.

The principle of this machine has been already explained (pp. 210—212), it therefore only remains, in this place, to indicate briefly the practical details and features of the construction. In this latter respect it is a much simpler machine than that above described; but still its efficiency depends in nearly an equal degree upon a correct appreciation of the principles involved in its *modus operandi*. The merely technical details have already been pretty fully pointed out in describing the plates enumerated above, but it may be necessary to indicate the rules employed in assimilating these to the conditions furnished by the particular circumstances of the individual case.

As in all other hydraulic machines, the data necessary to be assigned as the basis of any calculation of the size and angular velocity of the reaction wheel, are the values of H and Q, that is, the height of fall under which it is intended to act, and the volume of water to be used. We have before seen that if the water in the arms of the machine experienced no increase of pressure from centrifugal force, the discharge assigned by theory is expressed by $\mathrm{S}\sqrt{2\,g\,\mathrm{H}}$; but in consequence of the centrifugal force produced by the rotation of the machine about its axis, this quantity will be increased to $\mathrm{S}\sqrt{2\,g\ \mathrm{H} + v^2\left(1 - \frac{r^2}{\mathrm{R}^2}\right)}$. But we know from experiment that in consequence of frictional disturbance of the fluid in passing through the apparatus, the real quantity discharged is uniformly less than that assigned by theory, and that the reduction depends upon conditions which to some extent are within the control of the mechanician. On this subject we quote, with slight modification, from a paper read by Mr. W. M. Buchanan before the Philosophical Society of Glasgow (1846), on the theory of this species of machine. After stating the loss of head, observed in his experimental apparatus, by comparing the actual fall with the quantity of water actually discharged by a machine, of which the jet-orifices were accurately determined, the author assigns, as the sources of that reduction,

1° The pressure absorbed by the friction of the water in passing through the supply-pipe. This he regards as a known quantity, which is expressed in character and amount by

$$2f \cdot \frac{C}{A_{,}} L \cdot \frac{u^2}{2g},$$

in which C denotes the internal perimeter, $A_{,}$ the cross-sectional area, and L the length of the pipe; u the velocity with which the water descends through it, and f an empirical coefficient = .0035. If, therefore, S denote the sum of the areas of the orifices, V the velocity of efflux, and D the diameter of the pipe all in feet, this expression may be put under the form

$$8f \cdot \frac{L}{D} \cdot \frac{S^2}{A^2} \cdot \frac{V^2}{2g} = \alpha \frac{V^2}{2g}.$$

2° The loss of head arising from the acceleration of the water in passing from the supply-pipe into the interior of the machine through the water-joint neck, formed by the mouthpiece and central opening, and which is commonly less in diameter than the supply-pipe, as shown in Fig. 5 of Plate LXXXI. This he expresses by the formula

$$\frac{A_{,,}^2}{A_{,}^2} \left(\frac{1}{m} - 1\right)^2 \frac{\upsilon^2}{2g} = \beta \frac{V^2}{2g},$$

in which $A_{,,}$ is the area of the central opening, and υ the velocity of the water passing through it: m a coefficient determined from experiment to be = .9378.

3° The small loss of head resulting from the resistance encountered by the water in traversing the arms of the machine, which he expresses by

$$8f \,.\, S^2 \frac{V^2}{2g} \int_0^L \frac{C_{,}}{A_{,,,}} dx = \gamma \frac{V^2}{2g},$$

in which $C_{,}$ and $A_{,,,}$ are respectively the transverse perimeter and area of the channels at a distance x from their origin.

4° The loss resulting from what is called the *contracted vein.* Although the volume of water discharged by any orifice under a given head-pressure is invariably proportional to the area of that orifice and the square root of the head, the actual quantity is found to depend much upon the form of the orifice through which it issues. If the fluid be confined in a vessel of thin material, and the orifice be simply a hole pierced in its side, the discharge in cubic feet per second, will be nearly expressed by $\frac{5}{8}\, a \sqrt{2gH}$, the area of the orifice being a. If the jet from an orifice of this kind be closely observed, it will be perceived to converge through a short distance from its origin, forming, when the orifice is circular, a conoid, of which the area of the least section is $\frac{5}{8}$ of the area of the orifice. If advantage be taken of this circumstance to apply an ajutage to the orifice of the form assumed by the jet, the discharge will be found to approximate very closely to that assigned by the theoretical formula.

This difference of discharge in the two kinds of aperture is usually ascribed to the inclined directions which the molecules of the fluid assume previous to their exit, and which they tend to retain after passing the thin parietes of the simple orifice. For greater clearness let us assume that the aperture is horizontal, circular, and of small area in comparison with the area of the containing vessel; under these conditions a large portion of the fluid will be put in motion, and will slowly approach the orifice during the efflux, in the form of an inverted cone, of which the orifice is the apex. The particles, as they come opposite to the orifice, are therefore impressed with motions converging to an axis; but these motions, in consequence of the mutual cohesion of the particles, must tend to a common velocity in that axis: and the length of the external conoid will express the time in which the oblique motions are converted into motions parallel to the axis of the jet. It is therefore only at the point of least section that the molecules of fluid have attained the effective velocity due to the head under which they issue; and it is therefore only in reference to that point that the hydraulic pressure of the jet is equal to a column of the fluid of double the *actual head,* (p. 211). By adapting an ajutage to the orifice of the shape indicated, the oblique motions of the particles are corrected in passing through it, and reduced to parallelism with the axis at the moment of efflux into the atmosphere. There still, however, remains to depreciate the discharge assigned by the formula $q = a \sqrt{2g H}$, the imperfections of workmanship in the construction, and the adhesion of the fluid to the perimeter of the ajutage, with possibly a slight atmospheric influence not yet defined. But assuming the ajutage to be made with all possible care—both as to form and finish—if we call the area of the orifice 1000, that of the contracted vein will be 975: and these numbers taken inversely will express the velocity of the jet at the two points measured by the discharge. The value of q for an orifice of this form will therefore be

$$q = .975\, a \sqrt{2g H};$$

showing a loss of head-pressure, as measured by the discharge, of

$$(1 - .975^2)\frac{U^2}{2g} = .049375 \text{ H},$$

when $U = \sqrt{2g H}$ the theoretical velocity due to the head H. And generally, if V be the actual velocity of efflux, and k the practical coefficient of discharge for any orifice, so that $U = \frac{V}{k}$, the head-pressure *not realized* in the measure of q, will be $\left(\frac{1}{k^2} - 1\right)\frac{V^2}{2g} = \delta \frac{V^2}{2g}$. And the pressure *not realized* in the measure of the reaction, will be expressed by

$$\text{Sin}\, \phi \left(\frac{1}{k^2} - 1\right) \frac{V^2}{2g} = \delta_{\prime} \frac{V^2}{2g};$$

in which ϕ denotes the mean angle formed by the filaments of water of the jet with the axis.

But betwixt this the least contraction of the fluid vein, and that which takes place when the orifice is formed in a thin plate, we may evidently have a series of any number of terms expressing successive degrees of approximation of the ajutage to the theoretical form of least contraction. This is obvious, as regards the discharge from a fixed ajutage, and it is equally obvious, that if an ajutage be constructed to fulfil the conditions of least contraction when the vessel is at rest, it will no longer answer that condition when it moves in the line of the jet with any given velocity. If its motion be in the direction of the jet its length will manifestly be virtually increased, and the contraction will approach to that of a jet issuing from a parallel pipe, the coefficient for which is .8; and if the movement be in the contrary direction, the length of the ajutage will be in effect diminished, and the contraction will approach that from an orifice in a thin plate. This last is the actual case which falls to be considered in the reaction machine; the ajutages have a determinate velocity, in an opposite direction to that in which the fluid issues, and accordingly have their length virtually reduced. This must necessarily be provided against in the construction of the machine, and a length and form of the ajutages determined, which shall exactly correspond, at the given angular velocity of the machine, to the proper dimensions at which, if stationary, they would yield their maximum discharge. This is a problem which requires to be resolved for every machine.

It may, however, be stated as a general rule, that the contraction of the channels towards the orifices is half of that which would give the maximum discharge if the machine was at rest, and may therefore be taken at 7°.

If to these absorbing influences we add $\varepsilon \frac{V^2}{2g}$ comprehending the loss of atmospheric pressure due to the head H, and the effect of the cohesion of the water to the perimeter of the orifices (not valued) we shall have as the total calculated loss of head-pressure,

$$(\alpha + \beta + \gamma + \delta + \varepsilon) \frac{V^2}{2g};$$

and putting $\alpha + \beta + \gamma + \delta + \varepsilon = K$, we shall have as the velocity of efflux, taking the formula of p. 211,

$$\frac{1}{\sqrt{1+K}} \sqrt{2g\,H + v^2 \left(1 - \frac{r^2}{R^2}\right)}.$$

In those machines constructed according to the proportions usually adopted by the makers, the quantity $\frac{1}{\sqrt{1+K}}$ does not differ sensibly in ordinary cases from 0.94; and it has been stated (p. 69 of Description of Plates) that $R = 2.5\,r$; if therefore we substitute these numbers in this formula, it is reduced to the following:

$$0.94 \sqrt{2g\,H + .84\,v^2} = 7.5 \sqrt{H + \frac{1}{77}\,v^2} = V.$$

And multiplying this last expression by 60 times the area of the two orifices (in feet) we shall have, as the quantity of water discharged in a minute, $450\,S \sqrt{H + \frac{1}{77}\,v^2}$ cubic feet $= Q$.

In p. 212, we found as the measure of the effect of the machine $\frac{w}{g}(V - v)\,v$; if therefore in this expression we substitute the actual value of V found above, we shall have

$$E = \frac{w}{g}\left(7.5 \sqrt{H + \frac{1}{77}\,v^2} - v\right) v.$$

But, in practice, the velocity v of the machine is taken equal to $8 \sqrt{H}$. If, therefore, we substitute this value in that found for E, and put for w its equivalent 62.5 g, and for g its value 32.2, we shall have

$$E = 50\,q\,H \text{ very nearly.}$$

Or, taking the quantity of water expended in a minute, and expressing E in units of horse power, we have

$$E = \frac{Q\,H}{660},$$

which is this rule: Multiply the quantity of water expended in a minute by the given height of fall, and divide the product by 660: the resulting quotient will express the effect in units of horse power (the horse power being 33,000 lbs. raised through a height of 1 foot in a minute).

This rule shows that the machine ought to yield, in practice, an effect of 79½ per cent. of the power expended, independently of the partial losses of head above enumerated, taking the fall from the middle of the depth of the machine to the surface of the water in the reservoir. The following table of experiments, upon three model machines, will show that this high percentage of effect is attainable.

Height of fall H.	Quantity of water expended in 1 minute Q.	Weight on the arms of the friction brake p.	Velocity in the circle of the brake v.	Percentage of effect. $100 \frac{W\,H}{p\,v}$.	Diameter of model.
Feet.	Cubic feet.	Oz.	Feet.		
9.335	10.169	9	7910	75.212	7½ inches between centres of jets.
10.520	11.530	9¾	9510	76.664	
10.355	10.360	10¼	7820	74.933	
10.210	10.330	10	7820	74.433	
10.040	10.338	„	7900	76.333	
9.735	10.250	9¾	7820	76.630	
9.575	9.790	„	7330	76.460	
9.390	9.305	„	6690	74.868	
10.335	11.090	„	8960	76.440	
10.165	11.010	„	8840	77.236	
9.830	10.350	„	8030	77.173	
9.680	10.080	„	8420	77.887	
10.010	10.740	„	8580	78.040	
10.700	13.69	20½	5340	74.945	12 inches between centres of jets.
10.545	13.44	„	5240	76.015	
10.415	13.05	„	5040	76.237	
10.250	12.79	„	4900	76.845	
10.130	12.73	„	4870	77.644	
9.980	12.48	„	4630	76.425	
9.820	12.44	„	4500	75.745	
9.660	11.92	„	4280	76.420	
9.840	12.33	20¼	4600	77.000	
9.700	12.45	19¾	4750	77.910	
9.950	12.40	20½	4700	78.320	
10.270	12.68	21	4740	76.660	
10.460	12.47	22	4540	76.800	
	Lbs.				
8.05	562.32	14	3762.0	72.69	1.3 feet between centres of jets.
„	560.33	14½	3685.4	74.03	
„	558.75	15	3647.3	75.84	
„	553.61	15½	3488.5	76.01	
„	548.36	16	3380.4	76.56	
„	530.00	16½	3202.5	77.39	
„	514.25	17	2973.7	76.31	

The two models used in the first two sets of experiments were made by the writer, and the third by Mr. Whitelaw. In the experiments with the first models, the fall was variable, and the proper velocity of the machine was, therefore, in no case strictly attained. The correct velocity was not attained in the third set of experiments, although the fall was constant, in consequence of the successive variations of load being too great. The maximum of effect is, therefore, not obtained in any of the results given, but some of the results approach it very closely.

The mode of performing the experiments was nearly the same throughout. The load was applied upon the equal arms of a friction brake of 1.59155 feet radius (as nearly as could be measured), so that its circle was exactly 10 feet. The revolutions of the machine were ascertained by a counter worked by a screw cut on its vertical spindle; and the water discharged was received into a cistern, of which the dimensions were accurately determined. The circle of the arms of the brake at the points where the weight was attached being 10 feet, the numbers in the column stating the velocity in that circle being divided by 10, the quotient will, of course, show the number of revolutions made by the machine in the unit of time, 1 minute.

The constructive rules, published by Mr. Whitelaw in the *Artizan* for Nov. 1845, are as follows, the height of fall and the quantity of water furnished in a minute, being known.

A horse-power being taken at 33,000 lbs. raised one foot in a minute, this will be represented by 43,421 lbs. of water per minute, with a fall of one foot, supposing the machine to realize only 76 per cent. of the power expended; and the weight of a cubic feet of water being taken at 62.321 lbs., the equivalent of 43,421 lbs. will be 696.73 cubic feet. Taking Q and H as before, the quantity of water furnished in a minute and height through which it descends, we have as the value of E in units of horse power,

$$E = \frac{Q\,H}{696.73}.$$

From this the dimensions of the principal parts and the velocity of the machine, are determined as stated in the following expressions—it being understood that the machine has *two*, and only *two* jet orifices, and these so formed as not to cause the issuing jets to contract more than in the proportion of 97 to 100 after the fluid has left the orifices.

Width of each discharging orifice $= \sqrt{\frac{135\,E}{1000\,H\sqrt{H}}} = w_{,}$

Width of each arm of machine $= 4\,w_{,} = w_{,,}$

Diameter of the machine $= 50\,w_{,} = D_{,}$

Diameter of central opening $= 10\,w_{,} = d_{,}$

Number of revolutions in a minute $\frac{149.4338\sqrt{H}}{D_{,}}$

All these rules, except the last, may be departed from with impunity; but it is impossible to enumerate the circumstances and conditions under which modifications may be safely introduced, and where they would be prejudicial. These can only be appreciated by practice and a close investigation of the action of the machine. The rules are, however, safe within a wide range of fall—in fact for all ordinary cases.

SECTION IV.—COMPARISON OF THE DIFFERENT SPECIES OF WHEELS.

From what we have seen of the different conditions necessary to produce the maximum effect, it is evident that we ought not to be indifferent to the kind of wheel to be adopted in any particular case. The wheel ought especially to be adapted as far as possible, not only to the height of fall and quantity of water to be employed, but also to the kind of machinery which it is intended to propel. If the motion required be slow, and especially if it be besides irregular, a vertical wheel, of large diameter and considerable weight, will in general be the most satisfactory. On the contrary where a high velocity is required, a horizontal wheel will be the most economical. The undershot wheel is only commendable in cases where no other is applicable, on account of the lowness of the fall and large supply of water. It has the advantage, however, of being constructed at comparatively small cost, and if the run of the water be considerable, its velocity will be proportionally high in order that it may yield its maximum effect. Its great inconvenience is the smallness of that effect, which is in part remedied by employing the arrangement recommended by M. Poncelet now extensively adopted on the continent. This may be made to yield an effect of 40 to 50 per cent. of that of the water, when the head of water does not exceed 4½ feet. It also takes comparatively a high velocity, and it is to be kept in mind that the higher the velocity of any wheel of this species, the less will be its breadth, size of sluice, arc, and other parts influenced by the volume of water. It will, moreover, continue to work in back water until the levels before and behind approach equality, and is therefore particularly fitted for level districts subject to inundations. It is, however, liable to this inconvenience that its velocity cannot deviate sensibly from that at which it yields its maximum effect without losing greatly in power.

For falls from 4 to 7 feet, the breast wheel with radial floats inclosed in an arc may be employed with advantage. If well constructed, and the arc be accurately fitted, to prevent waste of water, this species of wheel is capable of yielding from 50 to 60 per cent. of the power of the water expended. It may besides deviate very considerably from the correct velocity without losing much in effect; but it is to be observed that this velocity ought never to be very high. This species of wheel is therefore particularly applicable in cases where the ultimate velocity of the machinery impelled is low; but it lies under the disadvantage that, on account of the slowness of its motion, its breadth must be great, and all its constructive details conformably large and heavy. It is, besides, not well suited for situations subject to back-water, which very speedily brings it to rest. The bucket wheel is applicable for higher falls and smaller supplies of water. It has all the advantages of the breast wheel, with some of its defects. It may have a velocity varying from 3 feet a second to 7 feet; and adopting this higher speed, and allowing the buckets to be half filled with water, its expense will be greatly lessened. When the fall is high, 18 feet and upwards, it is not common to provide any arc to economize the water at the lower part of the revolution; and when the buckets are properly formed, an arc is not greatly wanted. But in general cases the buckets are formed without much attention seemingly to the functions which they are intended to perform, and, accordingly, a large waste of power is incurred which an arc would go far to prevent.

The inconvenience of the bucket-wheel is the low rate of its maximum speed where high working velocities are required. This occasions the multiplication of intermediate geering with all its concomitant evils. It is applicable, we have said, to high falls; but this has its limits. To obtain the full value of the water, the diameter must increase as the fall, and the dimensions and weight assume corresponding proportions. The construction, accordingly, becomes expensive in a high degree.

The following table of the proportions of wheels of this class, constructed by Mr. William Fairbairn of Manchester, will be useful.

TABLE OF THE PROPORTIONS OF WATER-WHEELS CONSTRUCTED BY MR. WILLIAM FAIRBAIRN OF MANCHESTER.

	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.
Diameter of Water Wheel,	65 0	36 0	28 0	20 0	18 0	18 0	18 0	18 0	16 0	16 0	16 0	16 0	16 0	15 6	15 0
Breadth within the Buckets,	6 0	16 7	13 0	17 0	21 0	20 0	18 0	12 0	21 0	20 0	18 0	16 0	14 9½	17 6	6 0
Depth of the Buckets,	1 0	1 4	1 10	1 8	1 8	1 6	1 10	1 5	2 0	1 9	1 8	2 0	1 9	1 8	10
Revolutions of Wheel per minute,		1.95			4.78		6.15			7.8					
Speed of Periphery per second,		3 7		3.82	4 6		5 8			6 5	5 6		6.3	5.54	
Fall of Water,		33 2	26 6	16 6	13 3	16 0	10 0	16 0	9 0	7 10	9 6		8 0	8 0	14 6
Cubic feet of Water taken per second,				46		36		20	116		45				8
Diameter of Internal Driving Segment,	63 3	33 8		18 0	16 1		14 0⅝		14 0⅝	15 4¼	14 0	14 0	14 0¾		
Pitch of Teeth of Do.	3¼	3		3¼	3		3			3	3	3	3		
Breadth of Teeth,	10	1 2		1 0	1 2		1 0			8	1 0	1 0	9		
Estimated Horse Power,				60		52		30	70						12
		Tons cwt.qr.			Tons cwt.qr.				Tons cwt.qr.						
Actual Weight of Cast Iron,	...	32 7 0⅝	Men's time in the workshops occupied by this wheel, 185 weeks, and for erecting at the mill, 65. Total, 250 weeks.		26 14 3⅝	Men's time in the workshops, 123 weeks, and in erecting at the mill, 46 weeks. Total, 169 weeks.			23 15 1⅓	Men's time in the workshops, 102 weeks, erecting at the mill, 36 weeks. Total, 138 weeks.			Reckoned as one man's time.		
Do. Wrought Iron,	...	15 8 3¾			11 17 3⅓				8 15 1⅚						
Do. Brass,	...	2 2¾			1 3¾				1 2						
Total Weight of Wheel and Cast Iron Cistern,	...	47 18 3½			38 14 2¾				32 12 1½						

Note.—Mr. Fairbairn's maximum velocity of the periphery of the wheel is about 7 feet per second for a fall of 5 or 6 feet, and his minimum velocity from 2½ to 3 feet for a fall of 40 to 45 feet.

When the height of fall exceeds that for which it would be judicious to attempt to construct a bucket wheel, it is then necessary to have recourse to one or other of the horizontal wheels described—namely, the turbine of M. Fourneyron, or the reaction wheel of Mr. Whitelaw. The first of these may be made to yield an effect of about 50 per cent. of value of the water even on very low falls, and when immersed in tail-water; and about 70 per cent. on higher falls. The latter has also been made for situations in which it was intended to work occasionally in back-water, and with very considerable success; and, for high falls, we have seen, that it is capable of yielding an effect at least equal to the bucket wheel. It has advantage of the turbine in being less expensive, much more simple, and, we believe, is essentially more effective, on account of there being less loss of *vis viva* of the fluid in passing through the machine. Both have the advantage of being applicable to falls of any height, and of moving at velocities differing widely from that at which they yield their maximum, without much loss of effect. On low falls they have, what is generally reckoned, an advantage; they give immediately a comparatively high velocity which, in the case of cotton factories and the like, allows much heavy geering to be dispensed with. They have also a further very marked advantage in the little space which they occupy. Several reaction wheels, of 50 horse power and upwards, might be referred to, placed in situations where, it may be said, they literally occupy no room, being situated in a small pit under the floors of the factories to which they furnish the motive power. In cases where the fall is very high—and both the turbine and reaction wheel have been applied to falls which could not have been employed on wheels of the common kind—the high speed becomes an inconvenience, causing the use of intermediate geering for the purpose of reducing it. This can be remedied, to a certain extent, by increasing the diameter of the machine; but, when it is desired to take advantage of falls of 100 feet and upwards, perhaps some little inconvenience may be submitted to without reluctance.

ENGINEER AND MACHINIST'S ASSISTANT.

DESCRIPTIONS OF THE PLATES.

PLATES I., II.

BRACKETS AND PILLOW-BLOCKS.

THESE Plates contain views of a number of forms of brackets and pillow-blocks, adapted to various positions and purposes in mill work and engineering. The term *pillow-block*, *plummer-block* or *pedestal*, is applied to denote that part of a machine which forms the fixed point of support for the *journals*, or bearings of a revolving or reciprocating shaft; and the word *bracket*, although strictly applicable only to one particular form of framing, is frequently used to designate in general any piece of the fixed frame work which is employed for the purpose of supporting the pillow-block, so as to bring its axis into the proper position in relation to the other parts of the machine. The forms of these pieces are, of course, susceptible of infinite variety, according to the direction of the strain, the design of the adjacent parts of the framing, and the position of the axis in relation to them.

The design of the pillow-block itself is, in its main features, of a more uniform character, the principal difference in pieces of this kind being in regard to their size, which must, of course, be modified in proportion to the diameter of the shafts which they are intended to support. The ordinary form is represented in figs. 1, 2 and 3, which are an elevation, plan and transverse view of a pillow-block; these figures being each drawn partially in section, in order to make every part fully intelligible. The body *a*, and cover *b*, are usually made of cast-iron, and are carefully fitted together, so as to admit of being properly adjusted, without allowing any lateral motion. The *bushes c*, *c*, in which the journal works, and which are generally made of brass or gun metal, in order to diminish the friction, are also accurately fitted into the body and cover respectively; in the example now before us, their outer surfaces are of a polygonal form, to prevent their turning with the shaft, and they are provided with flanges to secure them from longitudinal displacement. The main object intended to be served by the introduction of the bushes is, to admit of the surfaces in immediate contact with the shaft being removed at pleasure, as they are worn out of shape by its friction. The bushes form together rather less than a complete cylinder, the deficiency being made up by slips of wood placed between their edges; between the body of the pillow-block and the cover, a corresponding space is also left and filled up in a similar manner. This arrangement allows of the accurate adjustment of the bushes; for when that which sustains the pressure of the shaft has been to a certain extent worn by the journal, the slips of wood may be removed and reduced in thickness, so that the two semi-cylindrical surfaces will be brought closer together, and embrace the journal more tightly. When properly adjusted, the body and cover are bound firmly together by bolts passing through both, the heads of the bolts being let into recesses cast on the under surface of the former, as shewn in fig. 1; and at this part it is expanded into a broad flat plate, called the *sole*, which, being accurately fitted to a corresponding part of the framing of the machine, affords the means of fixing them immovably together by bolts and keys. A small cup or chamber is usually cast upon the top of the cover, which is pierced by one or more small holes penetrating to the inner surface of the bush, for the purpose of admitting oil for the lubrication of the journal.

Fig. 4 is an elevation of a pedestal, suitable for supporting a shaft which requires to be raised to a considerable height above the base or sole on which it rests. Plummer-blocks of this kind are very commonly employed as the main-centre supports of the beam in steam engines; they are susceptible of considerable variety of design, according to the taste of the engineer.

The shafts of water wheels are usually supported upon pedestals of the form shewn in fig. 5, which also exhibits the mode of fixing such pieces of machinery to a foundation of masonry. Here the body of the pedestal is of the same form as that shewn in fig. 1; but as the strain is entirely in a downward direction, no cover is required further than to protect the journal from being injured by dirt, &c. Accordingly a false cover is usually provided, which consists simply of a thin shell of cast-iron, which, for the sake of appearance, is generally made of the same external form as in the ordinary plummer-block.

When a pedestal is to be supported upon a foundation of masonry, a sole plate *d*, *d*, is usually interposed; it is made of considerably greater dimensions than the sole of the plummer-block, and is bolted firmly to the surface of the building by long bolts embracing a great proportion of the stones of which it is composed. Upon this sole-plate two projecting pieces *e*, *e*, are cast, of a height corresponding to the thickness of the sole of the plummer-block, which is accurately fitted between them, sufficient space being left at either extremity to admit of dove-tail keys *f*, *f*, being introduced between the ends of the sole and the projections. The bolt holes in the sole of the plummer-block are made of an oblong form, so as to allow a certain amount of motion longitudinally, until the true position of the axis has been determined, when the keys *f*, *f*, are fitted, and driven firmly, into their places, binding the sole-plate and plummer-block into one compact mass. This arrangement is found to be of the utmost practical value and importance, and should be adopted in all cases where the lateral strain upon a pedestal is considerable.

In some machines it is necessary to provide the means of readily altering the position of a shaft or spindle with respect to the other parts. In these cases, some such contrivance as that represented in elevation and plan at fig. 13 is sometimes resorted to. Here the body of the plummer-block is in the form of an open rectangular frame, with a sole for securing it to its place as in ordinary cases. The interior surfaces and the sides of the frame are dressed truly parallel, and the bushes are fitted to slide within them freely, but without play. Two long pinching screws work through the ends of the frame opposite to the centre of the journal, their points bearing upon the back of each bush; these screws serve to adjust the axis to any required position, where it may be secured by means of the jam-nuts with which the screws are provided. One of the flanges of each bush is made of a separate piece from the body, and is attached to it by small screws; this is necessary in order to allow of their introduction within the frame.

Figs. 6, 7, 8 and 9 are specimens of hanging brackets, (or *gallowses*, as they are sometimes called), for carrying light shafts near the roof of a flat, to the beams of which they are attached by broad flat palms, as shewn in the figures. They are frequently provided with shells, or hollow covers, (see figs. 8 and 9), when the pressure is entirely downwards; but this should be avoided when the shafts are drivers acting upwards. The details of their construction, and the comparative advantages of the various kinds, are fully discussed in the Treatise on Geering.

Fig. 10 is a front view of an ordinary bracket, with pedestal attached. This form of bracket is useful for bolting to a wall, at the termination of a shaft.

Fig. 11 represents a common plummer-block fitted into a *wall-box* in the manner described in reference to fig. 5. The wall-box is simply an open rectangular frame of cast-iron, adapted to receive and secure the plummer-block, and built into the wall of the mill. Such an arrangement is indispensable in cases when a line of shafting is intended to pass *through* the wall; it is frequently also employed with advantage as the support for a shaft terminating at that point.

Fig. 12. A bracket supposed to be fixed to a wall or column; in this example the plummer-block is cast of the same piece with the bracket. When a line of light shafts is required to be carried parallel to a wall, and at no great distance from it, or through the middle of a flat where columns are also employed, such a mode of supporting them is usually adopted.

Fig. 14 represents an arrangement of columns, bridge, and pedestal, for supporting a horizontal shaft. This mode is sometimes adopted when it is requisite to carry a line of shafts into the middle of a flat, at a lower elevation than would conveniently admit of the employment of hanging brackets, and when the columns are necessary for supporting the floor above. This arrangement is sometimes adopted in flour mills; and occurs frequently in the framing of many kinds of machines.

Fig. 15 is an elevation of an arched bracket or *bridge* carrying an upright shaft. This arrangement is generally adopted when the upright shaft receives its motion from a *lying* or horizontal shaft by means of bevil wheels. The lying shaft passes underneath the upright shaft, and should have a bearing in a pillow-block keyed and bolted on the same bed-plate with the bridge, in order to secure the accurate and steady working of the bevil wheels. When the upright shaft is of any considerable size, the brasses guiding it laterally are made in two pieces to admit of their being tightened, and have a hollow in their upper edges to retain oil. The shaft is supported on a plate of brass or hard steel, called the *foot-step*. The box into which the brasses are fitted is cast open on one side, and provided with a door or cover having pinching screws passing through it to tighten the brasses. The foot-step is supported upon a block which may be removed, in order to admit of the shaft being lowered. These arrangements are more particularly illustrated, and their uses and advantages treated of, in the article on Geering.

Figs. 16 and 17 are front and edge views of a bracket adapted for carrying *guide-pulleys*. This sort of bracket, with its appendages, is usually termed a *universal guide*, and is employed for the purpose of changing the direction of a driving belt or strap when the pulleys over which it works are situated obliquely to each other, or when, from any cause, their connection cannot be directly effected. It will be observed that the circular bracket B is fixed to the beam D by two bolts which pass through circular slots cast in the palm of the bracket; so that the latter can be set to any required angle. The spindle of each guide-pulley works in studs, one of which is fitted to turn on the bracket as on a fixed centre, while the other passes through a circular slot, and is secured at the back by a nut. By this arrangement it is obvious that the guide-pulleys may be set to any required angle in either direction, so as to convey the motion to the proper point.

PLATES III., IV.

MISCELLANEOUS ENGINE DETAILS.

In these Plates we have grouped together a few examples of some of the principal working parts of steam engines of various kinds, with the view of familiarizing the eye of the young engineer with the most approved forms, proportions and modes of construction of which such parts are susceptible. The examples here given are for the most part taken from engines in actual operation; but no strict order has been observed in their arrangement, and the scales to which they are drawn are various.

Figs. 1 and 2 represent the ordinary form of cast-iron crank employed in land engines having wrought-iron crank shafts. The central *eye* of the crank is bored out truly cylindrical, to a diameter slightly less than that of the end of the shaft; it is then expanded by heating so as to admit of being slipped upon its place in the shaft, and on cooling, the contraction of the metal binds the two pieces inseparably together; they are usually further secured by riveting the end of the shaft. The crank pin is driven into its place with a sledge hammer, the surfaces of contact being made slightly conical, and is secured against relaxing by means of a key or pin passing transversely through it.

Fig. 3 is an elevation, and fig. 4 a section of an *eccentric*, with its rod and disengaging apparatus, such as is sometimes employed in marine engines. The eccentric A is a circular cast-iron disc, keyed upon the crank shaft, at a point more or less remote from the centre; the purpose for which it is most frequently employed in steam engines is to give motion to the valves which regulate the admission and escape of the steam. By the help of the turning lathe, a grove is formed upon its edge, which is embraced by the semi-circular pieces *a*, *a*, usually of gun-metal; these are screwed together at two points diametrically opposite, so as to form a ring, within which the eccentric revolves freely, but without play. The bolts which unite the parts of the ring together are usually prolongations of a forked rod B, B, destined to transmit the alternating rectilinear motion generated by the eccentric, to the pin *b*, of the lever which works the valves. But as it is frequently necessary to work the valves by hand, as at starting or stopping of the engine, the rod B is not jointed immovably to the pin *b*, but is capable of being readily disengaged from it, so as to allow it to move independently of the eccentric. For this purpose the socket *c*, in which the pin *b*, works, is made open on its under side, so that the rod B may be retained in geer with it only by its own weight; the end of a bent lever *d*, having its centre of motion in the upper surface of the rod B, is fitted to work through the latter at a point immediately over the socket or *gab c*, as indicated by the dot lines. The opposite extremity of the lever *d*, is provided with a handle, and towards that point also it is penetrated by a spring catch *e*, for the purpose of retaining it in its position when the pin *b*, is disengaged from the rod B, in which case the lower end of the lever *d*, is in the same line with the under surface of the eccentric rod. A light frame *f*, *f*, is generally added, by which the eccentric rod is guided between the collars of the pin *b*, and is thus prevented from being displaced while it remains out of geer with the valve lever. Thus, the disengaging geer being in the position indicated in the figure, it is obvious that the pin *b*, is at liberty to move in either direction, independently of the eccentric rod; but on pushing forward the catch *e*, the weight of the rod will cause it to fall into geer with the valve lever.

Fig. 5 is an elevation, and fig. 6 a transverse section of a highly elegant, compact and efficient form of the eccentric and its appendages, as adapted for a locomotive engine. In this case, the eccentric A itself is usually made in two pieces, for the convenience of fixing it upon the cranked axle of the engine. These pieces are carefully fitted together on a line passing through the centre of the axle, and are secured to each other by means of cotter bolts *g*, *g*. The eccentric is fixed in its proper position on the axle by the pointed set screws *h*, *h*, which admit of its being more nicely adjusted, and more readily altered as circumstances may require, than the method of keying. The bearing surface of the ring or strap *a*, *a*, is made concave, fitting accurately to the correspondingly formed edge of the eccentric, and a small groove is cut all round it, to admit of the free circulation of the oil which is applied for its lubrication by the oil cup *i*. The eccentric rod B is in this case bolted firmly to a palm formed on one of the semi-circular pieces *a*, *a*, in the manner indicated in the figure.

Fig. 7 shows the form of forked eccentric rod end, with which, until a comparatively recent period, locomotive engines were invariably provided. It was first employed by Messrs. Robert Stephenson & Co., and its introduction contributed not a little to the perfection and efficiency of the locomotive engine, although it has since been in most instances superseded by another contrivance, which shall be noticed presently. In order to explain the uses of these parts, it is necessary to observe that, in the ordinary locomotives, each engine is provided with two eccentrics, one for driving the engine forward and the other backward. By means of a combination of levers and rods, one of which is shown at *d*, *d*, either of these may be thrown into geer with the valve lever *b*, at the pleasure of the engine driver; and while either of the pairs is in its working position, as shown in fig. 7, the pin or stud of the lever traverses freely between the extremities of the forks *f*, *f*, of the other pair, without coming into contact with them; but when it is required to reverse the motion, the pair previously in geer are depressed, and the others being simultaneously raised, the pin *b*, is forced, in whatever position it may have been left, to slide along one of the inclined planes formed by the converging sides *f*, *f*, and to enter the notch or gab at the vertex, thereby causing the valve at once to assume and maintain the position proper for reversing the motion. Thus the use of starting levers is dispensed with, and the management of the locomotive engine is greatly simplified.

Figs. 8 and 9 represent a side and end view of Stephenson's patent *link motion*, by which the same object is attained, with the additional advantage of still greater simplicity, and of enabling the engine driver to vary the length of stroke of the valve, when it is required to work the engines expansively. Here B, B are the ends of the forward and backward eccentric rods for one engine; they are jointed respectively to the upper and lower extremities of the link C, C, which consists of two similar and parallel pieces of wrought-iron or steel securely bolted together, and curved to the circle which the eccentric rods would describe in their motion round the centres of their respective rings. On the interior surface of each piece, as indicated by the dotted lines, a groove is formed throughout its entire length, in which a rectangular block on the end of the valve spindle D is fitted to slide accurately, the valve spindle being guided in its rectilinear motion, by passing through a square socket, fixed to the frame of the engine. The link is suspended exactly at the middle of its length by means of a rod *d*, jointed to a pin projecting externally, and connected by a system of levers and rods to a handle situated within the control of the engine driver, and which admits of being readily secured in any required position.

From this description the action of this highly elegant and ingenious mechanism will be very obvious. Supposing it to be set into the position shown in the figures, that is, with the valve spindle D situated exactly midway between the ends of the eccentric rods B, B, little or no motion of the valve will ensue, as the link merely oscillates as upon a fixed centre at D. But if the link be raised or depressed, until the end of either of the rods B, B coincides with the position of the valve spindle, it is obvious that the full throw of the eccentric to which it belongs, will be imparted to the valve, and that in the most direct and simple manner, without the intervention of valve levers, weigh shafts, &c. Any amount of stroke, less than the full throw of the eccentric, may be given to the valve by adjusting the position of the link, so that its centre may be at a greater or less distance from that of the valve spindle.

Figs. 10 and 11. An elevation and section of an *axle box*, or bearing for one of the minor axles of a locomotive engine, showing the mode in which the spring is usually made to act upon it. E is the spring pin, passing downwards through the wooden framing of the engine, to which the ends of the spring are attached by short links; *j*, a cast-iron block bolted firmly between the plates in which the axle box works, and serving to guide the lower end of the pin E; *k*, the upper bearing of the axle, formed of gun metal, and fitted to slide vertically between planed guides, shown in dotted lines in fig. 10; the end of the pin E is received into a hole in its upper surface, and thus transmits the weight of the engine to the spring. The lower bush, which is introduced merely to protect the axle from dust, &c., is fitted between the projecting checks of the bush *k*, and secured to it by a single small pin or bolt; F is the longitudinal stay bolt, connecting all the axle guides of the engine.

Figs. 12, 13 and 14 represent an improved form of locomotive engine axle box, which has recently been proposed by Mr. Bodmer of Manchester. Its peculiarity consists in an ingenious application of the sphere and cylinder, constituting a species of universal joint, for the purpose of obviating any injurious strain to which the axle box may be subjected from the irregularity of the rails, or in passing round curves.

The outside framing of the engine G, G is bored on its interior surface to a truly cylindrical form, and the piece H, H is prepared in the turning lathe so as to fit it accurately; this piece supports the weight of the engine, being brought into contact with the spring spindle E, as in the example last described. The bush *k*, *k*, is of brass, cast in one piece, projecting considerably outside of the axle box, and bored internally to the exact form and dimensions of the axle. On its exterior surface is formed a spherical part, fitting to a corresponding concavity in the piece H; these spherical surfaces sustain the weight of the engine, a certain amount of space being left on all sides to allow of the free action of the bearing parts in obedience to the irregularities of the rails. A key *l*, is introduced into a hole in the interior surface of the piece H, and works in a rectangular hollow formed in the bush *k*, in order to prevent the latter from turning round. By these arrangements it is obvious that the axle will work freely in every imaginable position within the limits required in practice.

Numerous other applications of the same principles to locomotive purposes have been proposed by Mr. Bodmer. Of these we have given another example in fig. 15, which is a sectional plan of an apparatus contrived by him for combining the functions of the draw bar and feed pipe between the engine and tender. The construction is so obvious from the figure as to require no detailed explanation. Fig. 16 represents the ordinary draw link between the carriages of a train; it consists of a rod *m*, one-half of which is formed into a right, and the other into a left-handed screw, working into nuts, to which the connecting chains are attached by hooks. The weighted lever *n*, which is jointed to the middle of this rod, serves to draw the nuts together to the required extent, and to retain them invariably in the position into which they may be set.

The remaining figures in these Plates exhibit various modes of construction employed for connecting rods, pistons and piston rods. Figs. 17 and 18 represent a specimen of an ordinary land engine connecting rod, the length of which is determined by the distance between the centre of the crank shaft and that of the outer stud of the beam, when set horizontal. The material of which it is principally composed is cast-iron, strengthened throughout the greater part of its length by four ribs, or *feathers*, of a parabolic outline; the form of cross section at the middle is indicated at fig. 17. Its upper extremity is provided with two projecting branches, on which are fitted the wrought-iron straps I, I; these are firmly attached to the main body of the connecting rod by the cotters *o*, *o*. In the upper part of these straps are fitted the brass bushes by which the connection with the end of the beam is effected; they are adjusted to their respective studs by means of the cotters *p*, *p*. The lower end of the connecting rod is in a similar manner jointed to the crank pin; the ordinary method employed in the construction of this part is to make the lower end of the connecting rod solid, and to fit the brasses into it; an example of which form is given in Plates XIX.—XXII.; but the moveable wrought-iron strap J appears to us on many accounts preferable.

Figs. 19 and 20 represent the form of locomotive engine connecting rod recently adopted by Messrs. Robert Stephenson & Co. It is distinguished by extreme simplicity of construction, strength and security against derangement. The brasses *q*, *q*, for both ends are attached to their places by the strong double wrought-iron bolts *r*, *r*, which pass round their exterior surfaces, being lodged in suitable recesses formed for their reception, and are secured by nuts to the strong palms *s*, *s*, upon the ends of the rod K; these nuts serve also to adjust the brasses to the proper degree of friction upon their respective journals, and thus supersede the use of cotters, which are on many accounts objectionable in locomotive engine work.

Figs. 21 and 22 represent Mr. Bodmer's locomotive engine connecting rod, in which, as in so many of this gentleman's ingenious contrivances, he has applied the principle of the ball and socket universal joint. The crank axle end is formed by two semi-circular pieces L, L, to the innermost of which the rod K is strongly attached by a screw formed at its extremity; they are connected together and tightened up to the required degree, by means of two wrought-iron bolts *r*, *r*, passing through projections upon their outer surfaces. The inner surfaces of the semi-circular pieces L, L, and the exterior of the brasses *q*, *q*, for the reception of the crank pin, are turned respectively into a concave and convex form, so as exactly to fit each other, and thus admit of any slight deviation from the proper working position of the axle, due to the irregularities of the rails, &c., without straining the connecting rod. The opposite end is formed upon a similar principle, as indicated by the dotted lines, but the part in which the brasses are fitted is of a piece with the rod K and is close at the end, a pinching screw *t*, and steel block *u*, being employed to supply the place of a cotter.

Fig. 23 represents a specimen of a rod such as is used in a variety of circumstances in engine work, for conveying motion from one point to another. Sometimes, as in the case of parallel motion rods, the tightning cotters are placed on opposite sides of their respective bushes, in order that the length of the rod between the centres may not be affected by the wearing of the brasses.

In figs. 24, 25, 26 and 27, we have given two distinct varieties of pistons applicable to small high pressure engines, whether locomotive or stationary. The construction is so obvious from the figures as to render description unnecessary. P, P, are the pistons; R, R, the piston rods; *r*, *r*, the packing rings, formed in the one case of an equal thickness throughout, and in the other eccentric; S, S, the packing springs; *j*, *j*, the junk rings; *b*, *b*, bolts for holding down the junk ring; *f*, a dovetail ferrule for securing the piston rod in the first example, and *p*, *p*, conical pins for the same purpose in the other; C is the cross head surmounting the piston rod, for the convenience of attachment to the parallel motion links of the beam, or other mechanism by which the power is transmitted.

PLATES V., VI.

ILLUSTRATIONS OF BEVIL GEERING.

It is of the utmost importance to the young engineer to acquire a thorough knowledge of the mode of representing bevil wheels in all their various projections, not only on account of their frequent occurrence in machinery, but for the many interesting principles in mechanical drawing, and methods of proceeding applicable in all its departments, which are involved in their correct representation. Accordingly we have devoted these Plates to the illustration of this subject, and shall proceed to describe, in reference to them, the methods usually adopted in laying down such pieces of machinery.

Plate V. is occupied with projections of a bevil wheel of the kind technically called *meter wheels*. Such wheels are employed when it is required to transfer the motion of a shaft to another at right angles to the first, without altering the angular velocity—a case which occurs very frequently in practice. In a pair of wheels of this kind, the diameter, and consequently the angle of the teeth, in both being equal, it is obvious that only one pattern is required, and that the method of drawing one is applicable to both.

Fig. 1 is a section, fig. 2 a face view, and fig. 3 an edge view of a meter wheel. These projections are laid down in the proper relative positions for constructing the one from the other. The following is the process:—

Draw a line AB equal to the diameter of the wheel, (which is invariably estimated at the pitch circle); bisect AB in G, and through the point G draw a line GH at right angles to AB; the line GH will represent the centre line of the shaft. The points O and D in this line, (see figs. 1 and 3), where the axes of the two shafts could intersect each other, is the point to which all the lines running in the direction of the breadth of the teeth are drawn. The distance of the point D or O from the line AB is equal to half the diameter of the pitch circle. This will be evident, if from the point B a line BC be drawn at right angles and equal to AB; this line will represent the diameter and position of the other wheel, and if bisected in E, the line ED drawn perpendicular to it will be the centre line of the shaft. The point D is therefore the position of the common apex of the two cones.

Join AB and BD, and AO and BO, and at right angles to these lines draw *em* and *ri*, (see fig. 1), passing through the points A and B; these lines represent the *crown* of the wheel, which should be developed upon a plane surface, in the manner shewn to the right of fig. 3, in order that the form and dimensions of the teeth may be determined.* Make A*k*, B*n* equal to the breadth of the teeth, and draw *kg* and *nh* parallel to A*m*, B*i*. Set off upon A*m*, B*i* the length of the teeth, as *ef*, *rs*; also the thickness of the rim, as *fm*, *si*, and from these points let lines be drawn to the apex of the cones, as shown in the drawing. Join *gh*, and make *qv* equal to the size of the eye; *pq* and *vw* are each to be equal to the intended thickness of the eye, and *pt* equal to the depth. Draw *pt*, *qu*, *vx*, *wy*, at right angles to *gh*, and join *tm* and *yi*, and thus a section of the wheel is obtained, from which all the lines and points requisite for drawing the plan can be found.

In drawing the plan fig. 2, the mode of proceeding is to transfer the points 1, 2, 3, 4, 5, from fig. 1 to it by means of the T square; these lines determine the extremities of the teeth and rim. Circles are then described passing through these points, and that passing through the point 4 is divided into the number of teeth intended to be in the wheel; the thickness of the teeth at the roots and points is then set off, and joined by curves representing the projection of their outlines. The lines forming the edges of the top and bottom of the teeth are radial lines, and are, of course, drawn in the direction of the centre of the wheel.

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* See our Treatise on Geering, Section VII.

Fig. 3 is the side or edge view. The elementary lines for this drawing, viz. those marked 6, 7, 8, 9, 10, correspond to the lines in the section denoted by the same figures, and the points bounding the teeth are derived from fig. 2, by the application of the square. For example, the two points *a*, *b*, bounding the top of the outer extremity of the tooth F, are transferred to the line marked 8 on fig. 3, and the points *c*, *d*, to the line marked 7; these points being connected by the lines *ac* and *bd*, form the end of the tooth; straight lines are then drawn from *a*, *b*, in the direction of the point D, until they meet the line marked 10; also a line from *c*, tending to the same point and meeting the line marked 9. The intersections of these lines define the extremities of the inside of the tooth; and these being joined, the tooth F is completed. The outlines of the other teeth in the wheel are found by a similar operation to that just described, keeping in view that the slope of all the teeth tends to the point D.

The eye of this wheel is intended for a round shaft, on which it may be keyed in the usual manner. The thickness of the web is equal to that of a tooth of the wheel; the arms have a thickness somewhat less, and the thickness of the metal of the eye is equal to the pitch of the teeth.

The radius representing the circle of action of wheels of the kind described, is found by producing the line B*i* till it meets the centre line of the shaft.

In Plate VI. we have given examples of two pairs of bevil wheels, laid down with the view of illustrating the mode of drawing the teeth. They differ from the specimen represented in the preceding plate inasmuch as the wheels of each geering pair are of unequal sizes, and consequently effect a change in the velocity of rotation of their respective shafts; hence they are distinguished by the denomination of *bevil wheels and pinions*, the former term being applied to the larger and the latter to the smaller wheel of the pair.

The wheel and pinion represented in fig. 1 are both supposed to be placed on horizontal shafts, in which respect they differ from the pair shown at fig. 2, where the pinion is placed on an upright shaft, having its lower bearing in a foot step, while the wheel with which it is in geer is fixed on a horizontal shaft. The principle on which the projections of the teeth of these wheels are made is precisely the same as we have endeavoured to explain in reference to the preceding plate, as will be obvious from inspection of the drawings, and from the subjoined remarks.

Having determined the diameters of the pitch circles of the wheel and pinion respectively, in order that the required change in the velocity may be effected, let these lines be laid down meeting each other at right angles, as shewn at *c*, *c*, *c*, fig. 1. Bisect the diameter of the wheel in B, and that of the pinion in C, and draw the lines B O, C O, at right angles to the pitch lines and meeting each other in the point O; these lines represent the axes of the shafts of the wheel and pinion respectively, and the point O is the position of the apex of the two cones, to which all the lines bounding the sides and edges of the teeth tend. Then, (supposing that only the edge view of the wheels is required to be shown), construct, according to the method laid down in the preceding plate, so much of the section of the wheel and pinion as is necessary in order to determine the position of the lines *a*, *b*, *c*, *d*, *e*, *f*, and through the points *c*, *c*, *c*, draw straight lines at right angles to the sides of the generating cones; these lines will meet in two points situated in the axes of the wheel and pinion respectively, and to these points the lines bounding the outer ends of all the teeth in each are drawn.

In order to determine the position and dimensions of the teeth in this projection, a face view of the wheel is necessary, as was before explained; but this requires to be only partially laid down, as exemplified by the dotted lines in the figure. It is usual in drawing a pair of bevil wheels, to make the centre lines O B, O C to pass through the middle of a tooth; these lines therefore divide the wheel and pinion respectively into two symmetrical parts, consequently it is sufficient to project only one-fourth of the entire face view of each. But indeed, as it is only the exterior points of the teeth that have to be determined by this means, it will be found sufficiently exact for ordinary purposes merely to draw, from any convenient point in their respective axes, circles of the diameter of the wheel and pinion at the pitch lines and at the points of the teeth; and having divided them into the required number of teeth, as shewn by the numeral references on the figure, to mark off upon these circles the thickness at the point and pitch line. By applying the T square, these points are then transferred to their appropriate lines *c*, *c*, *d*, *d*, on the edge view, as denoted by the dotted lines *h*, *h*, *h*, *h*; from the middle of the teeth thus projected, straight lines are drawn to the point in the axis where the lines *d d*, *d d* meet, and the sides are formed by lines parallel to these centre lines, and tangents to the curves joining the thickness at the pitch line and points. The lines forming the edges of the teeth at the root and point, both in the wheel and pinion, are then drawn all radiating to the common centre O, as exemplified by the dot lines *g*, *g*; and being terminated at the lines *e*, *e*, and *f*, *f*, and joined by appropriate curves, complete the representation of this projection of the teeth. The pair of wheels represented at fig. 2 are drawn in precisely the same manner as above detailed, with this difference, that instead of being drawn parallel to centre lines, the straight lines *c*, *c*, forming the sides of the teeth, are all made to radiate to the points *a*, *a*, or apices of the cones defining the *crown* of the wheel; this facilitates the process considerably, and when, as in the present example, the wheels are drawn to a small scale, the deviation from strict accuracy will not be appreciable.

Example for Practice.—In order to attain to some degree of expertness in the representation of bevil wheels, the student of mechanical drawing should, after having made himself familiar with the methods above described, exercise himself in laying down wheels from such data as in the following example: Required to draw the section, plan, and side views of two bevil wheels, in the proportion of 3:2, the one being 4 feet 6 inches in diameter, containing 72 teeth, 7 inches broad; and the other being 3 feet diameter, having 48 teeth; the length of the teeth to be made equal to 7-10ths of the pitch, the thickness to 7-15ths of the same unit. The socket to be the same depth as the breadth of the teeth, the boss for the shaft 8 inches diameter, and the thickness of the socket equal to the pitch. The thickness of the rim at the outside to be 1¼ inch; each wheel to have six arms, 5 inches broad at the socket, 4 inches at the rim, and 7-8ths of an inch thick.

Bearings of Bevil Wheels.—As wheels ought never to overhang their bearings, when the contrary arrangement is possible, it would be necessary, in the example fig. 1, to contrive a double bracket or other fixing, which might be contained within the angle of the wheels, and at the same time have the bearings as near to the wheels as possible. This is often a problem to the millwright, involving much ingenuity in its proper solution. The pair shewn in fig. 2 affords an easy case of this problem; for, if the bottom of the upright shaft be supported upon a bridge of the same kind as that shown at fig. 15, Plate II., sufficient space can usually be obtained underneath the arch, and on the same base-plate which supports it, for a pedestal to carry the horizontal shaft. Thus the wheels are securely bound together, which is not the case either when they overhang their bearings, or have pedestals on separate parts of a wall or foundation. Indeed, in reference to this point, the principle involved in the above arrangement ought never to be lost sight of; namely, to confine the strain arising from the action of the wheels to their fixings, by connecting the bearings rigidly and inflexibly together. If this precaution be neglected, and the strain be referred to the wall and beams of the building, the geering will work less securely, and much greater strength of building will be necessary.

PLATE VII.

BEVIL MORTICE WHEEL AND PINION.

This Plate contains three views of a large bevil mortice wheel, geering with a small pinion; the ratio of their respective diameters being as four to one. This species of wheel differs from the ordinary kind in having wooden instead of cast-iron teeth; these teeth, which are usually termed *cogs*, are made of hardwood, and inserted into opening or *mortices* cast in the rims, into which they are secured either by dovetail keys *v*, *v*, as in the present example, or by iron pins passing through the cogs close under the rim of the wheel. Mortice wheels are employed with advantage, both in spur and bevil geering, where a high velocity, combined with great smoothness of action, is required. Their construction and uses are fully described in our Treatise on Geering, Sections I. and VIII.; which renders further detail in this place unnecessary.

Fig. 1 is a vertical section of the wheel, shewing the mode of laying down the elementary lines necessary for constructing the drawing of the wheel and pinion. This section also shews distinctly the form of the cogs, and the mode of their insertion into the rim. Fig. 2 is the plan or face view of the wheel, derived from the section in the manner indicated by the dotted lines to the left of the figures, and which in turn serves to determine the position of the teeth in the edge view fig. 3. In this last figure the pinion is also shewn in geer with the wheel.

The method of drawing the teeth of this pair of wheels is precisely the same as that which we have already pointed out in reference to the two preceding Plates. By comparing the dotted lines in fig. 1 with those laid down in Plate V. and following the directions given in the accompanying notice, it will be found that the point D is the apex of the two generating cones of the wheel and pinion, the bases of these cones A B, B C, being laid down respectively equal to the diameters of the pitch lines; to this point, therefore, the lines representing the face of the wheel and all those bounding the edges of the teeth are made to converge. The bevil of the ends, and the positions of those parts of the cogs which project through the back of the rim will be determined, as before, by finding the point in the axis at which straight lines drawn through the extremities of the pitch line, at right angles to the sides of the generating cones, intersect each other.

The manner of finding the points for drawing the plan fig. 2 is by applying the square to fig. 1, as shewn by the dotted lines 1, 2, 3, 4, 5, 6, 7, and the shape of the teeth in fig. 3 is precisely the same as described and illustrated in Plate V. The dotted lines *s*, *s*, *s*, *s*, figs. 2 and 3, shew the mode of transferring the extreme points of the teeth to the edge view.

PLATE VIII.

BEVIL AND SPUR WHEELS.

To complete our series of illustrations of geering, we present in this Plate an example of the combination of spur and bevil wheels, with their appropriate shafts and bearings. Such arrangements as the present are of frequent occurrence in practice, and the drawing will serve to shew the application of the various rules and methods we have pointed out to some of the most ordinary circumstances.

In this train of geering, the power is understood to be communicated from the prime mover by a shaft H, which may be supposed to be the crank shaft of a steam engine, or main shaft of a water wheel. Its extremity works in a plummer-block fixed to the sole *h*, which is securely bolted to a stone foundation, and close to this bearing a large spur wheel F is fixed to the shaft H by keys; this wheel works into the pinion G, which is fast upon the end of the horizontal shaft K. The plate *h*, is cast of sufficient dimensions to serve as a sole for the pedestals of both the shafts H and K, and the latter is supported at its opposite extremity by a similar plummer-block, bolted to a broad and strong sole plate *k*, *k*, which is common to all the geering deriving its motion from the shaft K. This geering consists of a bevil pinion

A fixed upon the vertical shaft V, working into the wheel C, upon the shaft K, and giving motion in its turn to a similar wheel B, upon the horizontal shaft L; the wheel E is upon the extreme end of this shaft, and geers with that marked D, which is supposed to be carried upon a cross shaft, and presents only a back view in this drawing; this last pair are of the kind termed meter wheels.

The lower end of the vertical shaft V works in a foot-step fitted into an arched framing or bridge *v*, *v*, of precisely the same form and dimensions as that represented at fig. 15, Plate II. and at fig. 2, Plate VI. It is firmly bolted in the centre of the sole plate *k*, *k*, and further secured by keys, as are also the inner bearings of the shafts K and L; by which arrangement the utmost compactness and stability is attained, and the masonry of the foundation in a great measure relieved from the strain of the geering.

To those who have followed us in our previous descriptions it will be unnecessary to enter into any detail regarding the mode of drawing the wheels of this train; we have, however, dotted in some of the principal elementary lines necessary for laying them down correctly. Thus the lines marked 1,1, 4,4, 7,7, define the boundaries of the generating cones for the faces and crowns of the various bevil wheels, and the points marked 5, 6, and 8 are the apices of these cones, to which all the lines forming the teeth are drawn, as exemplified at 2, 3.

PLATES IX., X.

WATER THRASHING MACHINE OF EIGHT HORSE POWER.

These Plates represent very clearly the general form and arrangement of the machine commonly employed in our farm yards for the purpose of separating the grain from the straw. The motive power for such machines is sometimes supplied by the strength of animals, sometimes by a water wheel, and occasionally by a steam engine; in the example now before us the prime mover is a water wheel of the kind usually denominated *overshot*. The only important modification of the machine itself as here represented, and which is sometimes adopted, consists in transmitting the power to the various subordinate moving parts by means of trains of toothed geering instead of by pitch chains; but the latter method appears in practice to be fully adequate to the end proposed, and renders the machine much less cumbrous and expensive. In some of the more modern thrashing mills also, cast-iron has been substituted for timber in the construction of the framing, but this is by no means general, and its superiority is questionable.

Plate IX. represents a side elevation of the machine, with the lining of the frame-work removed in order to expose the internal parts.

Plate X. is occupied with a general plan of the entire machine with the water wheel and geering.

General Description.

The framing of the machine A, A is composed entirely of timber, and is of the simplest construction, such as we might expect to find in localities where cast-iron framing would be less easily obtained. It is lined with thin deals so as to present the appearance of a close box, within which the working parts of the machine are enclosed. At one end is fixed a flat inclined board F, called the *feed table*, on which the sheaves are spread out in a thin layer previously to being introduced into the machine; this is effected by means of the two fluted rollers R, R working together close to the inner edge of the feed table, and moving at the rate of about sixty revolutions per minute. Immediately behind the feeding rollers, is situated the *drum* D, which is a hollow cylinder revolving horizontally at the rate of about 400 turns per minute, and carrying upon its periphery four projecting pieces *d*, *d*, called the *beaters*, their purpose being to thrash or *beat* the grain from the straw as it is brought under their action by the feeding rollers. The beaters are simply pieces of wood, disposed longitudinally, and at an equal distance from each other, upon the surface of the drum at right angles to its axis, and armed with plates of iron upon the sides which act upon the material. The bottom of the chamber within which the drum revolves is made in the form of a hollow cylindrical segment, with which the outer edges of the beaters move nearly in contact; and the top has a covering of wood of considerable strength, also close to the beaters, for the purpose of keeping the sheaves under their action.

Behind the beater cylinder D, and at a somewhat higher elevation, the *shakers* S, S revolve horizontally at the rate of about twenty-five turns per minute. These consist each of four leaves, or thin pieces of wood, strengthened by bars of the same material, and disposed in the form of a cross upon a central axis. Their outer extremities are each armed with a series of spikes, *s*, *s*, *s*, about six inches in length, set at a considerable inclination to the common axis. The shakers thus form a species of revolving rakes, and are employed for the purpose of tossing the straw to and fro in all directions, in order thoroughly to detach the grain from it. The points of the rakes revolve nearly in contact with the segmental bottoms H, H of their respective chambers; these consist of numerous slender ribs or spars, placed transversely, and at such distance apart as to allow the grain to fall freely through them into the subjacent hopper *k*, *k*, while the straw is tossed out behind the machine upon an inclined table or *heck* Y, by which it is conveyed into the straw barn. The hopper *k*, collects the grain from both sets of shakers, and conducts it to the fanners J, situated on the floor beneath the thrashing machine; here it is freed from dust, small fragments of straw, and other impurities that may have adhered to it, and will generally be found to be in a state fit for the market. In the more modern thrashing mills, self-acting *elevators* and *creepers** are sometimes employed to convey the cleansed grain as it is delivered from the fanners to the granaries or other proper receptacles where it is to be stored.

The Water Wheel and Geering.—The power requisite for giving motion to the thrashing mill, in the example now before us is derived, as has been before stated, from an overshot water wheel W W, the position of which in relation to the machine is clearly exhibited in Plate X. The form of the buckets *e*, *f*, *g*, *h*, *n*, which are of wood, is shown in dotted lines in the general elevation. The shaft *w*, on which it is hung, is of cast-iron, and rests on pillow-blocks bolted to the stone work forming the sides of the wheel-pit. These pillow-blocks have sometimes covers of the regular form, but more commonly they are unprovided with covers of any sort; occasionally they are fitted with *shell-covers*, (see fig. 5, Plate I), for the sake of appearance, and to preserve the bearings from water and sand. The wheel travels at the rate of about seven revolutions per minute.

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* For a detailed account of the nature and construction of these contrivances, see the description of Plates XCVIII.—CV.

On the end of the water wheel shaft which passes into the barn is keyed a spur wheel 1, 1, of 121 teeth; this wheel works into the pinion 3, of 16 teeth, upon the short intermediate shaft B, (see plan). This shaft carries also a spur mortice wheel 2, 2, of 115 teeth, geering with the small pinion 4, of 15 teeth, which is fast upon the axis of the beater cylinder D. The feeding rollers are also driven from the intermediate shaft B, by means of a pitch chain, passing over the pulley 8, upon the spindle of the lower roller. The shakers likewise derive their motion from a pitch pulley upon the shaft B, working by means of a pitch chain marked 7, into a similar pulley of considerably larger diameter marked 5, fixed upon the axis of the first set of shakers. The second set is connected with the first by a pitch chain marked 9, which passes over the two equal pulleys *a*, *a*, upon their respective shafts.

The fan pulley C is driven at a high velocity by a rope band from another pulley P of the same diameter, upon the axis of the beater cylinder or drum D; consequently the velocities of the driving and driven shafts would be invariably equal, supposing the tension of the band to remain always the same. But as it is necessary to modify the strength of the blast according to the quality of the grain, a simple contrivance is usually resorted to for the purpose of regulating the velocity of the fan. The driving band is made to pass over two small guide pulleys *b*, *b*, fixed to a frame which can be shifted vertically, so as to increase or diminish the tension of the band, and thereby allow of a certain amount of *slip*, according as the grain is heavy or light, a greater strength of blast being required in the former case than in the latter. The directions of the motions of the several parts of the machine are indicated by the arrows on the elevation drawing, and indeed, are obvious from the mode of its action, which will be sufficiently intelligible from the preceding description.

PLATES XI., XII.

IMPROVED FIVE ROLLER CALENDER. BY A. MORE AND SON, GLASGOW.

These drawings represent the form of Calendering Machine now generally employed in Glasgow for the finishing of certain descriptions of cloth. Such machines are mostly supplied, in that locality, by the above named firm, and their extensive practice in their peculiar department of mechanical operations is a sufficient warrant for the efficiency and excellence of the present arrangement. They have recently introduced an improvement in the mode of applying the pressure to the top roller; this mechanism is fully represented in our Plates, together with the system of geering employed for setting the machine in motion.

Enumeration of the Figures.

Fig. 1 is a front, and fig. 2 a side elevation of the whole machine with its driving and top pressure geering. The other figures show the constructive details of its various parts.

Fig. 3 is a plan of the entablature or framing which surmounts the machine, shewing the mode in which the top-pressure levers are applied.

Figs. 4, 5 and 6 represent the construction of the rollers.

Fig. 7. A sectional plan of the starting and reversing geering, and fig. 8 a transverse section of the side framing, through the bearings of one of the cast-iron rollers.

Fig. 9 shews the form of friction clutch employed for connecting the driving geering with the machine, and fig. 10 is a transverse section of the top pressure lever at the point where the elevating screw passes through it.

General Description.

The framing of the machine consists of two strong cast-iron standards or sides A, A, bolted to the floor, and securely bound together by stay bolts at the bottom, and at the top by a cast-iron entablature B. The side framings are traversed throughout the greater part of their height by parallel slots or openings, in which the cast-iron blocks for carrying the bearings of the rollers are fitted to slide vertically, while they are restricted from lateral motion; the pinching screws *a*, *a*, being provided for the purpose of adjustment. The rollers are five in number: the three great rollers C, C, C, being constructed of paper, in a manner which will be detailed below, while the two intermediate ones D, D, are cast-iron, made hollow to allow of the introduction of hot bolts of iron within them, or of steam when it is required or preferred. A series of rails, consisting of two fixed bars *b*, *b*, and the same number of rollers *c*, *c*, are mounted upon brackets bolted on front and towards the lower end of the framing; these are for the purpose of removing the creases, and putting the proper degree of tension upon the cloth in passing towards the lower roller. And when it has passed through the machine, and been subjected to its action until the requisite amount of finish has been attained, it is withdrawn from the upper roller by means of the stripping roller E, worked by a belt and pulley F from the driving shaft.

The machine is set in motion by a shaft G from the prime mover, lying in a direction at right angles to that of the rollers; a short intermediate shaft H, connected to the former by means of a system of reversing geer, serves to transmit the power to the machine itself. The reversing geer, which will be best understood by reference to fig. 7, consists of two bevil wheels I and J, fitted to turn loose upon the shaft, and both geering with the intermediate wheel K, upon the short cross shaft H. Between the wheels I and J, a double clutch or coupling box *d*, slides longitudinally upon the shaft G, and is constrained to turn with it by means of sunk feathers; it is commanded by a long lever-handle and clutch-fork *e*. The teeth of the clutch are fitted to engage with similar projections cast upon the inner surfaces of the wheels I and J, by which arrangement it is clear that the wheel K and short shaft H, to which it is fixed, will be driven in either direction according as the clutch *d*, is in geer with the wheel to the right or left. When set exactly midway between these wheels, the shaft G continues to revolve without moving the machine.

To obviate the risk of accident to the machinery arising from undue strain upon the rollers, a friction apparatus is interposed between them and the driving geering. This consists simply of two semi-circular plates of wrought-iron *f*, *f*, bolted together over the end of the shaft H, in such a manner as to be carried round with it only by the friction between the surfaces in contact, (see fig. 9); this friction may be regulated at pleasure by the connecting bolts, so as to admit of the shaft turning independently of the clutch when the resistance upon the latter exceeds a certain amount.

The friction clutch *f*, gives motion, by means of the projecting studs *g*, *g*, to the spur wheel L, which revolves loose in a bearing *h*, bolted to the framing of the machine. This wheel geers with a pinion M, fixed upon the end of the upper cast-iron roller D, which is connected with the lower roller by means of a system of spur wheels N, O, P, the relative sizes of which may be varied according to the nature of the work to be performed.

The requisite pressure is imparted, in the first instance, to the top paper roller C, and is by it transmitted, by direct contact of the peripheries, to the whole system. For this purpose the machine is surmounted by two strong cast-iron levers Q, Q, jointed to the entablature in the manner indicated in figs. 2 and 3. The opposite extremities of these levers are terminated by circular racks, or toothed segments R, R, securely bolted to the levers, and further strengthened by stay-rods. Opposite to the centre of the levers is situated a horizontal shaft, working in suitable bearings upon the upright standards S, S, and carrying two pinions T, T, geering with the circular racks. On the inner extremity of this shaft is also fixed a pulley U, to the rim of which is attached the end of a leather belt, having a weight W suspended at its opposite end. By this arrangement it is obvious that the gravitating tendency of the weight W, which may be increased or diminished as circumstances may require, is constantly exerted to depress the end of the levers Q, Q, and this pressure is communicated to the bearing of the top roller by the screwed pins *i*, *i*, the further uses of which shall presently be explained. After the required degree of finish has been imparted to the cloth, the levers are raised by hand, by means of the arrangement of mechanism shown in fig. 2. A hand-wheel *j*, *j*, is mounted upon a short axis, working in suitable bearings in the upright standard S; on this axis is also fixed a small bevil pinion *k*, geering with the wheel *l*, upon the lower end of a vertical shaft *m*, which carries at its upper extremity the pinion *n*, working into the bevil wheel *o*, on the axis of the spur pinions T, T. Upon the axis of the hand wheel is also fixed a ratchet-wheel *p*, into the teeth of which a pall is fitted to engage, for the purpose of retaining the levers in their elevated position.

The pins *i*, *i*, which convey the pressure of the levers Q, Q, to the top róller, are screwed at their upper ends and work through the nuts *q*, *q*, which are inserted into the levers in the manner indicated in fig. 10. They are turned by means of the hand wheels *r*, *r*, and at the lower extremities are connected by swivel joints to the cast-iron blocks *s*, *s*, which carry the bearings of the top roller. In this manner it is evident that the position of that roller, in relation to the levers, may be raised or depressed by turning the hand wheels *r*, *r*. But as it is necessary to raise *all* the rollers clear of each other when the machine is stopped, an arrangement has been adopted by which this is accomplished by the same motion. Two small round rods *t*, *t*, are attached to the block *s*, or seat of the top roller; upon these rods three cross pieces, or *bridges* of wrought-iron, are fitted to slide, and are secured by pinching screws when adjusted to their proper positions. The upper bridge is placed *half an inch* clear of the bearing of the upper cast-iron roller D, when all the rollers are in contact; the middle bridge is set *one inch* below the bearing of the middle paper roller C; and the lower bridge *an inch and half* below that of the lower cast-iron roller. By this arrangement it is evident that when the pressure screws *i*, *i*, are raised, the blocks of the top roller being attached to them, the rods *t*, *t*, are also lifted, and along with them the various rollers, as the bridges come successively into contact with their respective bearings. By this provision, the paper cylinders are preserved from being injured by the contact of the heated cast-iron rollers when the calender is working warm and requires to be stopped.

The construction of the paper rollers or cylinders is as follows: Upon each end of an arbor of malleable iron, of sufficient strength to withstand the necessary pressure without yielding, is fastened a strong plate of cast-iron *u*, (figs. 4 and 5) a little less than the diameter of the roller to be made; the plate is secured in its proper place by a ring of iron *v*, cut in two, and let into a groove or neck turned in the arbor. When the roller is finished, the annular pieces are kept in their groove by a hot hoop *w*, put upon the outside of them and allowed to cool. A plate is fitted on the other end, of exactly the same size, and in the same manner.

In building the rollers, one of the plates is taken off the arbor, but the other is allowed to remain in its place. The paper sheets of which the rollers are to be made, have each a circular hole cut in the centre of it, of exactly the same diameter as the arbor. The sheets are then put upon the arbor, and pressed hard against the fixed plate. When the arbor is filled with paper it is subjected to the action of a powerful Bramah press,—always adding more paper to make up the deficiency caused by the compression, until the mass is condensed to the utmost practicable limit. The half rings are then put in their place, to prevent the plate from being pressed back by the elasticity of the paper. The roller is now to be dried sufficiently in a stove, the heat of which causes the paper to contract so as to be quite loose. The roller is then again taken to the press, and the unfixed plate being removed, more paper is added, and the whole again compressed, until the roller is hard enough for the purpose to which it is to be applied. It is next turned truly in the lathe till it acquires a very smooth surface.

Action of the Machine.—The manner of passing the cloth through the calender varies very much, according to the amount of finish required upon it. The various methods are accomplished by different arrangements of the geering N, O, P, so that a calender calculated to do all the different kinds of finishing becomes a very complicated machine, on account of the quantity of geering required. For common finishing, the method of passing the cloth through the calender is as follows:—The cloth is passed alternately over the rails *b*, *b*, and under the rollers *c*, *c*, in order to remove the creases, and is then introduced between the two lower rollers; returns between the lower cast-iron and central paper rollers; passes again between the central paper and upper cast-iron cylinders; and is finally subjected to the pressure between this last and the top paper roller. Then, instead of being beamed off by the stripping roller, (as was the case previously to the introduction of the improved system of applying the pressure), the cloth is allowed to wind itself on to the top roller. During this process the levers Q, Q are raised, but the pressure is constantly maintained upon them through the intervening connections, by means of the weight W, which is wound up by the pulley U as the levers rise. When the piece of cloth has been entirely received upon the top roller, the action of the machine is continued until the amount of finish required is produced. The workman then stops the machine, and by means of the hand-wheel *j*, raises the lever Q, Q until the top roller is clear of the adjacent one; he then takes the end of the piece which has been wound on to the top roller, and passes it round the stripping roller E; then again setting the machine in motion the roller E winds off the cloth in a finished state, when it may be removed and its place supplied by another. The levers are then lowered to their original position by turning the hand wheel *j*, when the machine is again in readiness for finishing another piece.

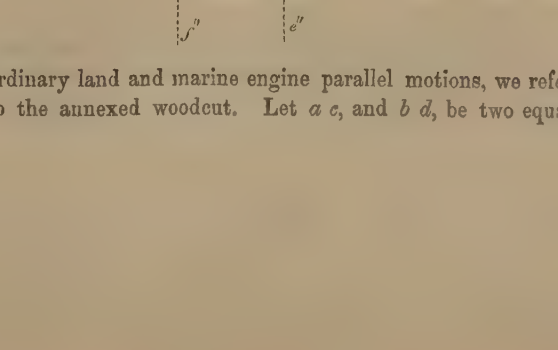
PLATES XIII., XIV.

PARALLEL MOTIONS AND GUIDES.

These Plates furnish examples of some of the most ordinary methods employed to maintain the rectilinear motion of the piston rod of a steam engine, under the constantly varying angular direction of the motion of the beam or connecting rod. Contrivances of this kind are required in many other similar cases besides that specified, but as it is absolutely essential to the economical working of the steam engine that the injurious oblique strain which would be produced under the circumstances referred to should be neutralized to the utmost extent practicable, more attention has been directed to the investigation of this subject than would otherwise, in all probability, have been deemed necessary. Accordingly, an almost endless series of contrivances, some of extreme elegance and ingenuity, have been from time to time produced under the name of parallel motions and parallel guides, the former consisting of various combinations of jointed levers and rods, and the latter of fixed and rigid rectilinear slides variously formed and applied, according to circumstances. Our Plates exhibit a few of the most remarkable of these combinations, shewn in conjunction with the parts whose movements they are designed to control and regulate.

The construction and mode of adjustment of the guide motions are sufficiently simple and obvious, and will require little further explanation than a mere reference to the circumstances under which the different varieties are applied; the subject of the parallel motions, on the other hand, strictly so called, is so curious and interesting that we propose, before entering upon the explanation of the Plates, to state concisely, and to illustrate in a simple and practical manner, the principles on which their action depends.

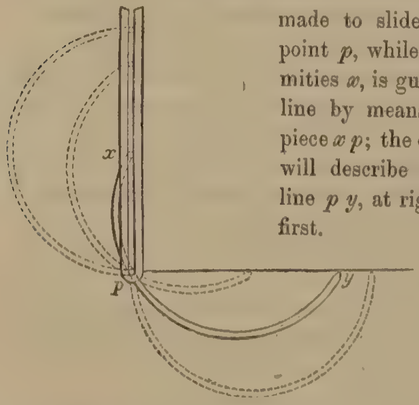
All the varieties and modifications of parallel motions, consisting of jointed levers and rods, hitherto proposed to be applied to the steam engine, depend for their action upon one general principle: that of counteracting the curvature of one circular motion by opposing to it a similar curvature operating in a contrary direction, in order that these may neutralize each other, and produce a rectilinear motion of the point to which the piston rod is attached. In practice this may be effected in two ways: 1st, by attaching the piston rod to a certain point in a link connecting the extremities of the levers which produce the counteracting motions, while their axes remain fixed; and 2ndly, by connecting the piston rod directly to the end of one of the levers, the axis of which shall slide horizontally in obedience to the curvilinear motion of the other. The first of these motions is only an approximation to mathematical accuracy, although sufficiently correct for all practical purposes, and is that which is most generally applied in the steam engine; the second is absolutely correct when the length of the levers is properly proportioned, but is objectionable in practice, for a reason which shall be stated when we come to consider its practical application.



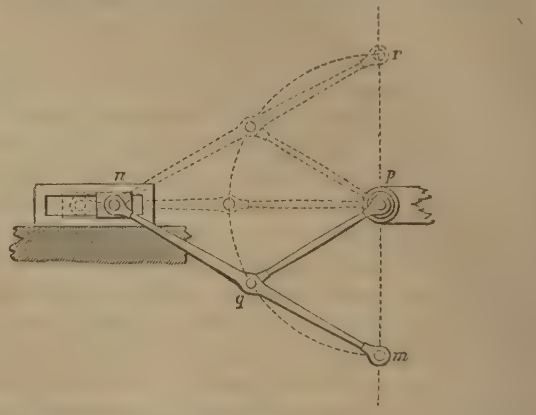
In illustration of the principle on which the first named combination is founded, and which forms the basis of the ordinary land and marine engine parallel motions, we refer to the annexed woodcut. Let *a c*, and *b d*, be two equal and parallel levers capable of moving freely round the fixed centres a and b; and $d\,c$, a link whose extremities are flexibly attached to those of the levers. Join $a\,b$ and $d\,c$, the diagonals of the parallelogram $a\,d\,b\,c$; the point of intersection e, divides each of these diagonals into two equal parts. If the levers $a\,c$ and $b\,d$ be now made to move round their fixed centres, so as to describe arcs of circles, the point e, will move in a line $e'\,e\,e''$, which will deviate more or less from a straight line according to the relative lengths of the levers $a\,c$, $b\,d$ and of the link $d\,c$.*

It may be easily demonstrated that if one of the levers, such as $b\,d$, be prolonged, and a point f, in the diagonal $a\,b$ connected with it and with the intermediate link $d\,c$, in such a manner as to form a parallelogram $f\,h\,d\,g$, the point f will, during the motion of the levers, describe a line $f'\,f\,f''$, parallel, and in all respects similar, to the line $e'\,e\,e''$. Hence, having secured the parallel motion of any point in such a system, it may readily be transferred to any other point by means of a parallelogram of jointed rods and levers. Such is, in fact, the ordinary land engine parallel motion, in which f, is the point of attachment of the piston rod, and e, that of the air-pump rod. In practice it is usual to make the lengths of $d\,g$, $d\,b$ and $a\,c$ equal, in which case the point f coincides with the point a, when the levers $a\,c$ and $b\,g$ are horizontal. This combination, as applied to the land engine, is exhibited at fig. 5, Plate XIII., as also at fig. 1, Plate XX.

The action of the second variety of parallel motions, which is also susceptible of many modifications, depends upon the well known mathematical principle that every angle which may be inscribed in a semi-circle is a right angle. In illustration of this principle, let a wire $x\,p\,y$ be bent into the form of a semi-circular arc, and made to slide against a fixed point p, while one of its extremities x, is guided in a straight line by means of the grooved piece $x\,p$; the other extremity y, will describe another straight line $p\,y$, at right angles to the first.



To exhibit the application of this principle in a practical form, suppose $m\,n$ to be a rigid bar having the end n, guided in a horizontal groove in which it can slide freely; and let $p\,g$ be also a rigid bar jointed to the former at q, and capable of moving round a fixed centre p. Let this bar be half the length of the bar $m\,n$, and let $m\,q = n\,q$; it is then evident from the principle stated above, that as the groove n, and the fixed centre at p, control the motion of the bar $m\,n$, the end m, is constrained to move in a straight line $m\,p\,r$, at right angles to $p\,n$; which is the condition to be fulfilled.



Another modification of this motion, as applied to a land engine is exhibited at fig. 3, Plate XIII. Here the main centre of the working beam, instead of being guided by a horizontal groove, as at n, is allowed to traverse a small arc which, deviating very little from a right line, fulfils the required conditions with considerable accuracy.

In the examples of parallel and guide motions given in our Plates, we have endeavoured to exhibit the forms and relative dimensions of the parts, as usually adopted in actual practice. Fig. 1 is an elevation of the beam or side lever of a steam engine, which, being fitted to work upon a fixed axis or *main centre* at A, receives an alternating motion from the piston rod. This motion causes the extremity B of the beam to move in a circular arc $a\,b$, and it is the object of the parallel motion to transmit to the point B the action of the piston rod, which moves in a straight line $c\,d$, bisecting the versed sine of the arc of vibration $a\,b$.

Fig. 2 is an example of the simplest form of the parallel motion, the construction being identical with that which we have chosen in demonstrating the principle. The form here exhibited may be applied to guide the motion of the piston rod of a direct action,† or crank over-head engine, where the crank is situated immediately over the piston rod; in such circumstances, however, guides are usually preferred. We have introduced this figure for the purpose of illustrating the mode usually resorted to in practice for finding the length and position of the fixed centre of the *radius bar*, or rod which is employed in every modification of the parallel motion to counteract the curvilinear motion of the beam.

Suppose the lever A B to represent part of the beam of a steam engine, whose extremity B, travelling in the circular arc $a\,b$, is jointed by means of a link L to the top of the piston or air pump rod P, and to the end of the radius bar R. It is obvious from the explanations already given that if the point of attachment e, of the rod and link be taken exactly in the middle of the latter, the length of the radius bar R will require to be made equal to the part of the beam A B in order that the piston rod may travel in the required path *c d*. If however the point *e*, be not situated exactly in the centre of the link, the length of the radius bar may be determined mechanically, as follows: the rod P with its attached link and beam, is made to move accurately in the line *c d;* the positions assumed during this motion by the lower end of the link are carefully marked at the three points *f*, *f*, *f*, and a circular arc *f f*, is described passing through these points; then the centre *g*, of that circle becomes the fixed axis of the bar R, and the radius *g f*, determines its length.

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* This deviation from the straight line is, as we have before remarked, exceedingly slight, and when the parts are properly proportioned and adjusted, is unworthy of being taken into account in practice.

† An example of this application of the parallel motion is given in Plates LXXV.—LXXVIII. of this work.

Fig. 3 represents a modification of the *second* variety of parallel motions, which is sometimes adopted in land engines of the smaller class. Here A B is the beam as before; the piston rod P is directly attached to its extremity B without the intervention of a link; the main centre A is capable of moving horizontally in a small arc of a circle, being supported upon a pillar, the lower end of which is jointed at C to the sole-plate. The radius bar R has its fixed centre at *g* attached to the framing of the engine, and is connected with the beam at a point *f*, exactly intermediate between the centre and extremity. The action of this parallel motion will be sufficiently obvious from the explanations already given; it is susceptible of considerable accuracy, but is liable to the grave practical objection, that the vibrations of the pillar on which the main centre is supported greatly impair the stability and durability of the engine, and introduce a needless complication of parts.

Fig. 4 represents a variety of the ordinary land engine parallel motion distinguished for its extreme simplicity of construction; it is applicable in cases where no air-pump is employed. The piston rod P is attached to the lower end *e*, of a rigid link L, which is jointed to the beam at B, somewhere about the middle of its length, while its upper extremity *f*, is connected with the radius bar R, moving upon a fixed centre at *g*, the position of which, as well as the length of the bar R may be ascertained in the manner described in reference to fig. 2. By this arrangement, the lower end *e* of the link will move in a line *c d*, which approaches nearly to a straight line, because the circular arc *a b*, described by the point B, is counteracted by an arc of smaller radius described by the bar R, in such a manner as almost to neutralize its effect at the point *e*.

Fig. 5 is an example of an ordinary complete land engine parallel motion and beam of approved form and proportions, as usually applied in practice. We have already had occasion, in the course of this investigation, to point out the principles on which the action of this elegant combination of mechanism depends, and it is unnecessary here to recapitulate them. A B B′ is the beam, L the back link or strap, in which the parallel motion is in the first instance produced, and to the centre of which the air-pump rod P is jointed. R is the radius bar, working upon a fixed centre at *g*, in the framing of the engine; *p*, the parallel bar for transferring the parallel motion of the link L to L′, the main or piston rod link; P′ the piston rod jointed to the latter at *e′*. It is usual to make the lengths A B, B B′ and *g f*, equal to each other, but should circumstances not admit of this arrangement, the length of the radius bar may be determined mechanically by marking a number of points *f*, *f*, *f*, when the rod P is moving parallel, and finding their common centre, as before explained. The links and bars of this, as of most other parallel motions, are made double, and disposed symmetrically on each side of the beam, while the piston and air-pump rods are attached to intermediate cross-heads connecting both sides.

Fig. 6 is the form of parallel motion usually employed in marine engines, but like the preceding, it is susceptible of various modifications. Its action is based upon the same principles with that of the motion represented in fig. 5, and the adjustment of the length of the parts may be effected by the methods already pointed out; it is in fact simply the land engine parallel motion inverted, and otherwise modified to suit the circumstances in which it is placed. The same letters of reference have been employed to denote the corresponding parts and positions in each, which will render a detailed description in the present case unnecessary. The radius lever R is usually actuated by one only of the side levers A B′; consequently its centre of motion becomes a cross shaft extending between the side framings of the engine, and provided with similar levers *e g*, at each extremity. The link L′ in this case represents the side rods, and L the rod which transfers the action of the beam to the radius lever shaft. To compensate for the wear of bearings and other causes of derangement, the rod L is usually made capable of adjustment longitudinally, and also with reference to its point of attachment to the side lever; an example of which mechanism will be found fully detailed in plates LVI.—LXIII.; the position of the cross shaft *g*, also sometimes admits of adjustment. The object of these various arrangements is to cause the rod *p*, to move truly parallel to the side lever in all positions of the stroke, which would obviously produce the required rectilinear motion of the point *e′;* it must, however, be remarked that the parallelism is not absolutely correct; but is true only within certain, though narrow limits, giving an approximation sufficiently near for ordinary practice.

The last example of parallel motions, strictly so called, which we have given in these Plates is of a somewhat peculiar, though extremely simple kind; it is applicable to engines in which a fixed point of support for the centre of motion of a radius bar cannot readily be obtained, and where no air-pump is required. This modification is represented in fig. 7, and will be at once understood on inspection of the drawing. Here A B is the beam, L the link, and P the piston rod as before. Upon the axis of the beam is mounted an adjustable toothed segment R, geering with another segment R′ of smaller diameter, and capable of turning upon an axis which is bent into the form of a crank. To this crank is jointed the parallel rod *p*, the opposite extremity of which is attached to the piston rod and link L, at the point *e*. Thus, during the stroke of the beam, the segment R′ is made to move through a space which, measured upon the pitch-line, is exactly equal to that through which the beam itself travels; consequently the extremity *f*, of the parallel bar will assume the positions *f*, *f*, when the beam is at the top and bottom of the stroke. It will readily be perceived that in these, and where, after a great many laborious attempts, they did make the engine work; but not being either philosophers to understand the reason, or mathematicians enough to calculate the powers and proportions of the parts, they very luckily, by accident, found what they sought for.

"They were at a loss about the pumps, but being so near Birmingham, and having the assistance of so many admirable and ingenious workmen, they came, about 1712, to the method of making the pump valves, clacks, and buckets, whereas they had but an imperfect notion of them before. One thing is very remarkable: as they at first were working, they were surprised to see the engine go several strokes, and very quick together, when, after a search, they found a hole in the piston, which let the cold water in to condense the steam in the inside of the cylinder, whereas, before, they had always done it on the outside. They used before to work with a buoy to the cylinder, enclosed in a pipe, which buoy rose when the steam was strong, and opened the injection and made a stroke; thereby they were capable of giving only six, eight, or ten strokes in a minute, till a boy named Humphrey Potter, in 1713, who attended the engine, added (what he called *scoggan*) a catch that the beam always opened, and then it would go fifteen or sixteen strokes a minute. But this being perplexed with catches and strings, Mr. Henry Beighton, in an engine he had built at Newcastle-upon-Tyne, in 1718, took them all away but the beam itself, and supplied them in a much better manner."

The next improvement to the engine was the addition of the cold water pump, worked from the reciprocating beam, to raise the water required for injection above the level of the cylinder. This water, after having done its duty in the cylinder, was expelled, along with the condensed steam, through an escape-valve, into the feeding cistern, by the descent of the piston.

A.D., 1718.—Beighton's improvements on the atmospheric engine date from the year 1718, from which time this engine continued in use, with but little alteration or improvement, for half a century. Brindley, indeed, applied an improvement to the feeding apparatus, which he made self-acting, by means of a float on the surface of the water in the boiler, communicating by a wire with a valve in the feed-pipe, which was thus opened whenever the water in the boiler fell below a certain level.

SMEATON, A.D., 1767.—In the year 1767, Smeaton's attention was first directed to the steam engine from having been employed to construct an atmospheric engine for the New River Company. Being of opinion that power was lost by the too rapid reciprocation of the beam and the other moving parts of the engine, since the whole momentum acquired during the stroke was necessarily annihilated at each change of direction in the motion, he resolved to increase the length of stroke, and remove the centre of the working beam nearer the pump-rod end, employing larger pumps to compensate for the slowness of the motion, and loading the piston with 10½ lbs. per square inch in place of 7 lbs., which had been the usual practice. One of the chief advantages he contemplated from these alterations was a diminution in the amount of injection water required to condense the steam, but in this he was disappointed; finding, to his surprise and mortification, that more injection water was necessary than before, and that the cylinder was consequently so much cooled, that much useful effect was lost by the condensation of the steam on first entering it. He accordingly altered the fulcrum of the beam until the load upon the piston was reduced to 8¼ lbs. per square inch. "With this load the engine went so much quicker, as not only to raise more water but consume less coals; it took less injection water, the cylinder became hot, and the injection water came out at 180° Fahr., and the engine, in every respect, not only did its work better, but went more pleasantly."

With the view of pursuing these investigations, he erected a small experimental engine, of about one horse power, at Austhorpe, from which he obtained the following results:—the consumption of coals was 55 lbs. per hour, equivalent to a duty of about 2,920,000 lbs. raised one foot high by a bushel of coals—the injection water required was 95 cubic inches per stroke, and the water evaporated from the boiler about 8·9 cubic inches per stroke. He came to the conclusion that the most beneficial load on the piston was 7·8 lbs. per square inch. Smeaton ascertained about this time the performance of most of the atmospheric engines then in use, and of fifteen at Newcastle the best was found to raise about seven and a half millions of pounds one foot high by the consumption of a bushel of coals, and the worst three and a fourth millions. The engine he afterwards erected at Long Benton, in which his improvements were fully carried out, raised nine and a half millions.

From what we have said it will appear that Smeaton introduced no material innovation in the principles of the atmospheric engine, his improvements being confined chiefly to the introduction of more accurate workmanship and juster proportions than heretofore; by which means alone he was enabled to effect a saving of probably 1⅓ in the consumption of fuel. He adopted the plan of clothing his cylinder and piston with a non-conducting substance, to check the radiation of heat, and he enlarged the steam-passages to the cylinder, by which the "throttling" of the steam was prevented during its passage from the boiler. His boilers were constructed on sounder principles, and were calculated with a much greater evaporative power, than any which had been employed before his time. They were of cast iron plates, bolted together by flanges, and were usually of the "haystack" form, hemispherical at top, and arched internally at the bottom; the flame being conducted, by the aid of flues in the setting, along the bottom and round the sides and ends, as in our modern boilers of this kind. Wrought iron plates were not used in the construction of boilers until a much more recent period, cast iron and copper having been the materials employed; and in several of Newcomen's old engines, the dome of the copper boiler was sometimes formed of lead.

The speed of Smeaton's engines was regulated by the contrivance called a cataract, the parent of the modern mechanism of the same name, and which had been applied to the mines in Cornwall some time previously. The apparatus is very simple, consisting in a small vessel balanced on a centre, so that when empty it stands erect, but when filled with water by the dripping of a small cock, it is over-balanced, and in its fall raises a lever, which opens the injection cock, and causes the engine to make a stroke. The speed of the engine is thus determined by the number of times that the vessel fills and empties itself per minute. In estimating the value of Mr. Smeaton's improvements, we should not forget that several of the mechanical inventions which he first applied to the air-pump, were adapted by subsequent engineers to the steam engine,—such as the close cylinder-cover and stuffing-box, &c.

James Watt, A.D. 1763.—In the year 1763, James Watt, then a philosophical instrument maker in Glasgow, first devoted the powers of his master-mind to the study and improvement of the steam engine, which thenceforth rapidly progressed under his guidance to a state of perfection little short of what we find it at the present day. His improvements merit our admiration the more that they were, for the most part, elicited by a patient and progressive course of reasoning, submitted for verification to the test of practice; and they exhibit, in a remarkable degree, the rare combination of true science with that practical and persevering turn of mind, which is requisite for adapting the abstruse theories of the philosopher to their noblest use —the service and amelioration of mankind.

Separate Condensation.— Watt's attention was first directed to the steam engine from being employed to repair a working model of a Newcomen engine, for the natural philosophy class of the Glasgow university. His experiments proved that the quantity of steam consumed by this engine was vastly more than what was effectively used in the cylinder. This loss he attributed partly to the small size of the engine, in consequence of which the cylinder presented a large comparative surface for radiation; but more especially to the useless condensation of the steam by contact with the sides of the cylinder, cooled down as it was by the injection water at each stroke. He therefore conceived the happy idea of condensing the steam in a *separate vessel*, which could always be kept cold without prejudice to the temperature of the cylinder. He surrounded the latter, at the same time, with a hollow cast iron casing, or "jacket," into which steam was suffered to flow from the boiler. The condenser, on the other hand, was kept as cold as possible, being generally immersed in a cistern of cold water.

The Air-Pump.—This arrangement suggested the adoption of the *air-pump* to withdraw from the condenser the injection water, with its contained air, the water of the condensed steam, the air from leakage, and the uncondensed watery vapour, all of which had been formerly expelled through the snifting-valve, by the descent of the piston. In Watt's earlier engines he employed condensation by surface, in order to save the work of the air-pump in lifting the injection water, but this was afterwards abandoned for the better method of condensation by injection. Dr. Roebuck, of the Carron iron-works, having heard of Watt's improvements, offered to join him in a patent, on condition of being allowed two-thirds of the profits. This was willingly agreed to, and in the year 1769, a patent right was applied for and obtained for fourteen years. The specification of this patent, taken out in Watt's name, was as follows:—

"My method of lessening the consumption of steam, and consequently of fuel, in fire engines, consists of the following principles:—

"First, That vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in common fire engines, and which I call the steam-vessel, must, during the whole time the engine is at work, be kept as hot as the steam that enters it; first, by enclosing it in a case of wood, or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and, thirdly, by suffering neither water or any other substance colder than the steam to enter or touch it during that time.

"Secondly, In engines that are to be worked wholly or partially by the condensation of steam, the steam is to be condensed in vessels distinct from the steam-vessels, or cylinders, although occasionally communicating with them; these vessels I call condensers; and, whilst the engines are working, these condensers ought at least to be kept as cold as the air in the neighbourhood of the engines, by application of water or other cold bodies.

"Thirdly, Whatever air or other elastic vapour is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the steam-vessels, or condensers, by means of pumps, wrought by the engines themselves, or otherwise.

"Fourthly, I intend, in many cases, to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the same manner as the pressure of the atmosphere is now employed in common fire engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only, by discharging the steam into the open air after it has done its office.

"Lastly, Instead of using water to render the piston, or other parts of the engines, air and steam-tight, I employ oils, wax, resinous bodies, fat of animals, quicksilver, and other metals in their fluid state.

"And the said James Watt, by a memorandum added to the said specification, declared that he did not intend that anything in the fourth article should be understood to extend to any engine where the water to be raised enters the steam-vessel itself, or any vessel having an open communication with it."

Partnership with Boulton, A.D. 1773.—Much difficulty was experienced at first from the impossibility of procuring accurate workmanship, and up to the year 1773, little progress had been made in the manufacture of the engines. In that year Dr. Roebuck's affairs became embarrassed, and he sold his interest in the patent-right to Mr. Boulton, of Soho, Birmingham, who at that time possessed the most celebrated metal works in the kingdom. An extended term for the patent was then applied for, and obtained, for twenty-five years from that date. Watt had now at his command

PLATES XVII., XVIII.

HIGH-PRESSURE PILLAR STEAM ENGINE OF TWELVE HORSE POWER, BY R. SANDERSON AND CO., GLASGOW.

The form of steam engine represented in these Plates is a species which has recently come greatly into vogue among steam engine makers. The distinctive feature of this kind of engine is the single column employed to support the main centre L of the beam. The axis of the column is exactly under the centre of the beam, but the uppermost member of the capital is cast of an oblong shape, which enables it to serve as a platform upon which the main centre pedestals are bolted down. To assist the bolts of these pedestals in withstanding the upward strains to which at intervals they are subjected, their soles are provided with dovetail recesses into which correspondingly formed projections, cast on the upper surface of the plinth, enter, leaving sufficient space on each side for fixing keys to be driven in, an arrangement which is fully shown in fig. 1, which is a side elevation of the engine; fig. 2 being an end view.

In this form of engine the column performs the whole work usually devolved upon a framing of four or six columns with entablature; and the single column having thus to receive and sustain both longitudinal and lateral strains, is consequently of a massive and substantial character, and communicates to the engine a compact and solid appearance. But we must not expect to find in constructions of this kind the graceful proportions which characterize the classic column; the material and the purposes to be served are different, and produce in the mind ideas so dissimilar that the proportions which please in the temple appear ludicrous in the engine. Indeed, the attempts we so frequently observe at imitation of the architectural forms of antiquity in structures of cast-iron are rarely successful, and it would, perhaps, be more judicious to abandon altogether the classic models than to press them into use under circumstances of contrast to which they are not adapted without modifications which render them displeasing, in consequence of the manifest and unsuccessful imitation which they evince.

The cylindrical form is not the best adapted to withstand the strains to which the column is liable, and accordingly it is in some instances connected by a bracket to the cylinder C, as in marine engines. This bracket, besides strengthening considerably the whole frame work, becomes available for carrying the fixed centres of the parallel motion, and likewise for supporting the upper end of the governor spindle. When the bracket is omitted, as in some inferior examples, these centres must be sustained by other means; sometimes they are carried upon a short bracket erected on the capital of the column, but more commonly on the main centre pedestals. The governor spindle is supported in the same manner.

The sole of the engine A, A is firmly bolted to a stone foundation by long bolts, which pass through strong washer-plates in the lower course of the building, and are usually secured by cotters, but sometimes also by nuts, as at their upper ends. All the parts are firmly bolted down upon the sole-plate, thereby giving to the engine a degree of portability and solidity which cannot easily be attained with other forms of framing.

The steam is admitted to the valve-casing B by the steam-pipe D, and is regulated by the throttle valve *i*, connected by means of the lever *h*, *h*, with the governor, which is driven from the crank-shaft through a train of light shafts and meter wheels marked G, *j*, *k*, *l*. The valve is worked from the traverse shaft *e*, *e*, by the side rods *b*, *b*; these rods are jointed at their upper extremities to the cross-head *a*, to which the valve rod is rigidly attached by nuts, to admit of its accurate adjustment; and their lower ends are connected to the short levers *c*, *c*, projecting from the traverse shaft *e*, *e*. This shaft is terminated in a lever *d*, at right angles to the former, and which carries a stud fitted to engage with the gab of the eccentric rod F, thereby forming a joint which admits of the eccentric rod being lifted out of geer by a lever handle *f*, when the engine is to be stopped. The other end of the rod F is fixed to the ring which embraces the eccentric *g*, on the crank-shaft, and is thereby made to travel longitudinally, at every half revolution of the crank-shaft, through a space equal to the throw of the eccentric. This motion is transmitted by the combination of levers above detailed to the cross-head *a* of the valve, which thus acquires a vertical motion, corresponding to the throw of the eccentric and relative lengths of the levers *d*, and *c*. In the present example these are equal, and consequently the travel of the valve is simply equal to the throw of the eccentric.

The feed pump H is worked directly from the beam of the engine K, K by the rod *n*. This rod is jointed to the solid plunger *o*, which works in a stuffing-box bolted on the top of the pump-barrel. It is usual, in high-pressure engines, to heat the feed water partially before forcing it into the boiler; for which purpose the waste steam ejected by the pipe E is made to play upon its surface; *p*, is the pipe communicating with the well or cistern containing the feed water, and *q* the branch which conducts it to the boiler.

The species of parallel motion here represented is a modification of Watt's original motion. The radius rod has its fixed centre *t*, in the bracket *z*, which is bolted to the diagonal stay passing between the main centre column and the steam cylinder. It is connected to the beam K, K by the links *s*, *s*, and to the cross-head of the piston rod *v*, by the parallel bars *r*, *r*. The cross-head is fixed on the top of the piston rod, by means of a cotter and gibs, which pass through it and the end of the piston rod; and the extremities of the arms are carried in bearings formed by the strap links M, M. These are close at the upper end and open below, in order to allow of the introduction of the brasses for the cross-head and stud in the end of the beam. The top brasses of this last being fitted into their places, the under ones are supported against the centre pins by two stay pillars, which rest on the top brasses of the cross-head joints. These under joints are completed by fitting in the lower brasses, and locking them in their places by cotters and gibs *u*, *u*, which pass through both legs of the straps underneath the brasses. The bearings for the upper and lower ends of the connecting rod are adjusted by means of the cotters *w*, and *x*, in the usual way.

PLATES XIX.—XXII.

CONDENSING ENGINE OF THIRTY HORSE POWER, BY NEILSON AND MITCHELL, GLASGOW.

These Plates represent very fully the form and constructive details of a very excellent specimen of that class of beam steam engines to which it belongs. The distinguishing feature of this description of engine consists in the main centre or axis of the working beam being supported upon a strong framing composed of a sole-plate, six cast-iron columns, and an entablature. These engines have a very light and handsome appearance, and moreover possess the maximum capability of resisting severe working strains; and although the cost of their original production be somewhat greater than that of engines of a simpler construction, they are invariably found to be the most economical in point of durability and expense of maintenance for repair.

These drawings are carefully laid down to the scales respectively appended to them, and, for further practical utility, the dimensions of the principal parts are marked in figures in the detail Plates XXI. and XXII.

Enumeration of the Figures.

Plates XIX., XX. Fig. 1 is a side elevation of the engine, with a portion of one of the columns at the steam cylinder end removed, in order to allow the internal arrangements of the cylinder and nozzles to be shewn in section. The section is taken in the plane of the axis of the cylinder, and is continued through the air-pump, condenser and hot water pump.

Fig. 2 is a general end elevation of the engine, taken at the cylinder end.

Plate XXI. Fig. 3 comprehends both a plan and elevation of part of the sole-plate towards the crank end, shewing the mode in which the crank-shaft plummer-block is formed in the body of the sole-plate.

Fig. 4. A plan and transverse section of the entablature, with the tops of the columns shewn in their position.

Figs. 5 and 6. An elevation and sectional plan of the steam cylinder and nozzles.

Figs. 7, 8 and 9. A face view, external and sectional plan of the steam valve.

Fig. 10. Sectional plan of the cold water cistern, with the condenser and air-pump shewn in their proper positions.

Fig. 11. An enlarged section of part of the air-pump, shewing the bucket with its valve, and the discharging valve also in section.

Figs. 12 and 13 are front and edge views of the crank of the engine.

Fig. 14. Two views of the connecting rod detached.

Plate XXII. Fig. 15 is an enlarged general view of the parallel motion spread out for the purpose of exhibiting all the parts in combination.

Fig. 16. A front elevation and longitudinal section of one of the main links of the parallel motion.

Fig. 17. The cross-head of the piston-rod.

Fig. 18. The cross bar for connecting the lower ends of the back links.

Fig. 19. Front elevation and longitudinal section of one of the back links of the parallel motion.

Figs. 20 and 21 represent the parallel and radius bars respectively.

Fig. 22. The cross-head of the air-pump rod.

Figs. 23 and 24 are the stud-centres of the air-pump and piston rods respectively.

Fig. 25. Front and edge view of the wrought-iron links of the connecting rod.

General Description.

The Framing.—The sole-plate A, A is cast in a single piece, and firmly bolted to a suitable foundation of masonry, within which the condensing apparatus is enclosed. It is provided with six circular recesses which are accurately bored out, and fitted to receive the turned ends of the columns B, B, B. The entablature C, C consists likewise of a single casting, with corresponding recesses bored to the diameter of the projecting ends of the columns, which are firmly secured at both extremities by strong cotters. The main centre pedestals D, D are bolted to the entablature immediately above the middle columns. The crank-shaft revolves in two plummer-blocks E, E, one of which is formed in the sole-plate, as shewn at fig. 3, while the other is supported upon a wall of solid masonry situated beyond the fly wheel. The governor works in a small frame F, bolted to the foundation of the engine, over-arching the crank-shaft and terminating in a hollow fluted column, within which the spindle is enclosed.

The Cylinder, Nozzles and Condensing Apparatus.—The steam cylinder G, which is firmly bolted to a suitable platform cast upon the sole-plate at one of its extremities, consists, as usual, of a separate casting from the valve casing H, which is joined to the cylinder with red lead cement, both faces having been previously planed and fitted together. Upon the back of the valve casing is bolted the steam pipe I for conveying the steam from the boiler to the engine; the throttle valve *a*, which is enclosed within the steam pipe, serving to regulate the quantity of steam admitted into the cylinder, and consequently the velocity of the engine; for which purpose it may be acted upon either by hand or by the engine itself, through the agency of the governor. Here is also situated the blow-through valve *b*, for opening a communication between the boiler and the condenser previous to the starting of the engine; it is enclosed within a small chest formed upon the lower of the two packing port covers *c*, *c*, which may be removed at pleasure to admit of the hemp packings at the back of the valve being examined and renewed when necessary. The steam valve *d*, *d*, is of the kind denominated *long-D-valves*, and is worked up and down against the face of the cylinder by means of a mechanism to be hereafter described, in such a manner as alternately to admit the steam from the boiler into the cylinder above and below the piston, through the upper and lower steam ports *e*, *e*. The bent pipe J, usually called the *exhaust pipe*, forms the communication between the valve casing and the condenser K, which is a close cast-iron vessel, destined for the production of a vacuum by the condensation of the steam.

The cold water cistern N is placed, as usual, in the masonry of the foundation, and contains the condenser K and air-pump M. These communicate by the passage L, in which is the foot valve *f*, opening towards the air-pump. The condenser is slightly conical in shape, and the upper and narrower end is provided with an internal projecting collar, for the purpose of receiving the extremity of the exhaust pipe J, which being made tight with hemp packing, thus forms an expansion joint of a very simple, though perfectly efficient description. The lower end of the condenser projects slightly into the passage L, through a hole formed in the upper surface of the latter, and is supported upon it by a flange cast round the condenser for that purpose. The air-pump is attached in a similar manner to the chamber L, at the other side of the foot valve, but projects downwards considerably more, for the purpose of giving the pump a better hold of the water. The valve of the air-pump bucket *g*, is of the perpendicular-lift, or *pot-lid* kind, as is also the discharging valve *h*. The water which is raised by the air-pump is discharged into the hot well O, from which it is withdrawn by a pipe *i*, to a reservoir or other receptacle, a portion of it, however, being intercepted for the supply of the boilers. This is effected by means of the feed pump P, which communicates with the hot well by the horizontal pipe *j*, and discharges into the boiler by the supply pipe *k*. The water in the cistern N is maintained at a uniform level and low temperature by means of the pump Q, which communicates with the cistern by the pipe *m*. The plungers of both hot and cold water pumps receive their motion from one common centre in the working beam.

The Working Parts.—The power generated within the steam cylinder G is impressed upon the piston *n*, and transferred by the piston rod *o*, to the beam R through the intervention of the parallel motion *p*, *p*. The details of the parallel motion are very minutely represented in Plate XXII., and the theoretical principles of its action having already been fully discussed in a former part of this work, we need not in this place revert to the subject. The beam, oscillating upon its axis at *q*, transmits the motion of the piston, by means of the connecting rod S to the crank T upon the end of the crank shaft U, the fly-wheel V serving to equalize and regulate the velocity. The air pump bucket and feed and cold water pump plungers partake of the motion of the beam, being connected to it by the rods *r*, *r*, and *s*, *s*, respectively.

The steam valve *d*, is worked from beneath the casing by means of an ingenious combination of mechanism, remarkable for its simplicity and perfect efficiency. An eccentric *t*, fixed upon the crank shaft U, communicates an alternating rotatory motion to the rocking shaft *v*, by means of the eccentric rod *u*, which is strongly, but lightly, framed together of wrought-iron rods. From the extremities of the valve levers on the rocking shaft are suspended two links, which are jointed by brasses and cotters to two projecting pins cast upon the outer surface of the sliding stuffing box *w*. The valve spindle is firmly attached to the lower end of this stuffing box, and passes upwards through the interior of a cast-iron tube bolted to the bottom of the valve casing, the outer surface of the tube being turned truly cylindrical, and fitted to work steam-tight through the stuffing box *w*. By these arrangements it is obvious that the valve spindle will be made to move truly parallel, and that any leakage which may occur, through wear and tear of the packing, may be remedied with the utmost facility by means of the stuffing box cover. The rod *x*, seen in fig. 1, is for the purpose of suspending a counterbalance weight to facilitate the working of the valve.

The Governor.—The governor W is mounted upon a vertical spindle revolving in bearings within the column F, and driven by a pair of bevil wheels *y*, from the fly-wheel shaft. The upper part of this spindle is hollow, and, at that part upon which the sliding collar works, is slotted through to allow of a connection between the collar and the rod which passes down through it. The divergence or collapse of the balls, as the engine moves with greater or less velocity, produces a vertical motion of this rod, which, by means of a system of small rods and levers *z*, *z*, is conveyed to the throttle valve, and thus the speed of the engine is regulated.

Action of the Engine.—To exemplify the action of this engine, let it be supposed that the throttle valve *a*, is open, and occupies a position parallel with the axis of the steam pipe I, thereby allowing a passage for the steam from the boiler to the valve casing; although the piston is, in the present instance, at half stroke, still, on account of the *lap* of the valve, the steam cannot enter the cylinder to start the engine, for both of the steam ports *e*, *e*, are shut, as represented in the section. But the steam being thus admitted into the valve casing, if the blow-through-valve *b*, be now opened, the steam will immediately flow into the condenser, and opening the foot valve *f*, will expel the air with which we may suppose it to be filled, through the air pump bucket and delivery valves. When the air is expelled—which is known to be the case as soon as pure steam is seen to issue from a small valve, called the snifting valve—the blow-through-valve is shut, and the injection cock opened to admit a shower of cold water from the cistern into the condenser, which, immediately condensing the steam contained in the latter and in the passage L, forms a vacuum in them. The rocking shaft *v*, must now be moved by a portable lever which fits into a socket on the shaft, (seen in fig. 2), until the pin in the fork of the working lever comes beneath the gab of the eccentric rod *u*; this motion will place the slide valve *d* in the proper position for opening a communication between the valve casing and the lower part of the cylinder, and between the upper part and the condenser. The atmospheric pressure being thus in great measure removed by the expansion of the air above the piston, into the condenser, and the steam being made to act upon its lower surface, motion of the piston in an upward direction will ensue, which will be transmitted through the piston rod *o*, to the beam R, and thence by the connecting rod S to the crank T, upon the fly-wheel shaft U. Motion being thus induced, when the piston arrives at the end of its stroke, the momentum of the fly-wheel will carry the crank over the lower centre, by which time the eccentric will have reversed the position of the slide valve; a communication will now be opened between the condenser and the lower part of the cylinder, and between the boiler and the upper end, thereby producing a downward motion of the piston, and consequently continuous revolution of the fly-wheel shaft. The injection water, together with that resulting from the condensation of the steam, meantime flows along the passage L, through the foot valve *f*, and is pumped away by the air pump M, into the hot-well O, from which it may either be allowed to run to waste or to flow back to the cold water reservoir.

Literal References.

A, A, the sole-plate of the engine.
B, B, B, the columns for supporting
C, C, the entablature.
D, D, the main centre plummer-blocks.
E, E, the crank-shaft plummer-blocks.
F, the framing or support of the governor.
G, the steam cylinder.
H, the steam valve casing.
I, the steam pipe from the boiler.
a, the throttle valve.
b, the blow-through valve.
c, *c*, the packing ports of the steam valve.
d, *d*, the steam valve.
e, *e*, the upper and lower steam ports.
J, the exhaust or eduction pipe.
K, the condenser.
L, sole-plate, forming also the passage between the condenser and
M, the air-pump.
f, the foot valve.
g, the air-pump bucket.
h, the discharging valve.
N, the cold water cistern.
O, the hot well.
i, the discharging pipe.
j, pipe between the hot well and
P, the feed-pump.
k, branch from the feed pump to the boilers.
Q, the cold water pump for feeding the condenser cistern N.
m, pipe for conveying the water to the cistern.
n, the piston.
o, the piston rod.
p, *p*, *p*, the parallel motion.
R, R, the beam.
q, *q*, the main centres of the beam.
S, S, the connecting rod.
T, the crank.
U, the crank-shaft.
V, V, the fly-wheel.
r, *r*, the air-pump rod.
s, *s*, the rods for working the feed and cold water pumps.
t, the eccentric.
u, the eccentric rod.
v, the traverse, or rocking shaft.
w, movable stuffing box, to which the steam valve rod is attached.
x, back balance rod.
W, the governor.
y, pair of bevil wheels for driving the governor.
z, *z*, system of rods and levers for conveying its action to the throttle valve.

PLATES XXIII.—XXV.

BOILERS FOR A CONDENSING ENGINE OF FORTY HORSE POWER, BY ROTHWELL, HICK AND ROTHWELL, BOLTON.

The important functions now performed by steam in most departments of the mechanical arts—whether as a source of motive power, or as a convenient means of dispensing heat—forbid the omission, in our collection of Plates, of the apparatus employed in its production; this apparatus forms, in fact, an indispensable adjunct to the numerous examples of steam engines given in this work.

It is absolutely necessary for the makers and proprietors of boilers to determine the amount of surface required to produce a certain quantity of steam in a given time, and the dimensions and disposition of the fire-grate, flues and chimney, in order to obtain that result with the least possible expenditure of fuel. Into these questions our limits do not admit of our entering in this place:* the Plates now before us furnish, however, a good example, which may serve as a guide, in these respects, for the construction of others. They represent a pair of low-pressure wagon-shaped boilers, of the construction introduced by Watt, and since his time so extensively in use; they are fitted with all the most approved and perfect feeding and safety apparatus, &c., and may be used either conjointly or separately, being each capable of supplying steam for a twenty horse engine. The detail sheet, Plate XXV., shows the more important parts of the apparatus referred to, on an enlarged scale.

Enumeration of the Figures.

Plates XXIII., XXIV.—Fig. 1 combines an end elevation of one of the boilers, and a transverse section of the other immediately in front of the feed pipe, both being shown in their proper relative positions, with all the necessary pipes for withdrawing the steam and feeding the boilers shown in external elevation. Also, a transverse section of the boiler-house.

Fig. 2 is a longitudinal section of one of the boilers, taken on the centre line, and exhibiting the internal arrangements of the boiler and flues.

Fig. 3. A vertical section, on a greatly reduced scale, of the chimney in which the flues of both boilers terminate.

Plate XXV.—Fig. 4. A vertical section of the feed pipe and its adjuncts. Fig. 5 is also a vertical section of the cast-iron float or bucket working within the feed pipe.

Fig. 6. A plan of the small cistern surmounting the feed-pipe.

Figs. 7 and 8. Section through the axis, and general plan of the steam pipes for connecting the two boilers and for leading the steam to the engine. In these views the construction of the safety and stop valves is fully exhibited.

Fig. 9. A longitudinal section of one of the stop valve chests, with its appendages.

Fig. 10. A plan of the cover of the spherical safety valve chest, as seen from below, showing the form and disposition of the ribs which serve to retain the ball in its place.

Fig. 11. An external side elevation of the conical safety valve chest, with its appendages.

Fig. 12. A section and plan of the cast-iron man-hole chest, showing the mode of inserting the bolts for securing the door or cover.

Fig. 13. Details of the apparatus for indicating the height of water in the boiler.

Fig. 14. Two views of one of the furnace bars.

Fig. 15. Two similar projections of one of the supports for the furnace bars.

Fig. 16. Section of an inaccessible safety valve, such as is frequently employed in place of the ball safety valve, represented in fig. 7.

Fig. 17. Another form of safety valve, having the weight accessible.

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* Much valuable practical information on these points will be found in Armstrong's excellent "Essay on Steam Engine Boilers," London, 1839.

General Description.

The boilers A, A are composed of an assemblage of wrought-iron plates, of the form and dimensions indicated at figs. 1 and 2; the plates which are exposed to the action of the fire being slightly thicker than those of which the dome or crown of the boiler is composed. The plates are strongly riveted together so as to be perfectly water-tight, and at once to guard against collapse, and to enable the boilers the better to withstand the united pressures of steam and water to which they are exposed, their sides are bound together internally by numerous cross stay rods *a*, *a*, *a*, which are secured thereto by strong cotters. They are each set in a foundation of brick-work, properly prepared for its reception, having suitable chambers for the admission of the fire grate, and for the passage of the current of flame and heated air under and round the boiler into the chimney. These, and all the other products of the combustion, after having traversed the channel B under the boiler, are diverted by means of the cross flue C, at its further extremity, into the great lateral flue D, and, passing along the front of the boiler through the chamber E, return down the second lateral flue F, and are then delivered into the chimney G. This circulation of the heated gases round the boiler has the effect of abstracting from them the greater portion of their caloric, and transmitting it to the water; care must, however, be taken that this principle be not carried to excess, as too low a temperature in the chimney would be highly prejudicial to the action of the boiler by injuring the draught. At fig. 1 is shown the cast-iron furnace door *b*, mounted upon its frame, also of cast-iron, and the flue doors *c*, *c*, which may be opened when required, to admit of the flues being cleaned. The form and dimensions of the furnace bars *t*, *t*, and of their transverse supports, are given at figs. 14 and 15, Plate XXV.

The steam which is generated within the boiler accumulates in the upper part, and by reason of its elasticity, tends to escape by any aperture which may be presented. It is retained until required to be brought into action, by means of the valve *n*, enclosed within the chest H, called the *stop valve.* When this valve is opened the steam is conducted along the pipes I and I′ to the cylinder of the engine, if it is to be employed as a motive power, or to its destined point of action, whatever object it may be required to serve.

In order to prevent any danger that may be apprehended from the undue accumulation of elastic force within the boiler, an inaccessible safety valve *p*, loaded with a weight equivalent to the greatest pressure to which that force is to be limited, is enclosed within the chest J, fitted to the pipe I, and allows the surplus steam to escape by the pipe L, as soon as it has attained that pressure. The peculiar kind of valve here represented in our Plates will be more fully described below. Another safety valve *o*, enclosed within the chest K, is also fixed upon the steam pipe, and the excess of steam which passes through it escapes by the pipe L. The weight upon this latter valve, which is of the conical kind, is somewhat less than that with which the valve *p* is loaded, and it is brought, by means of a combination of levers and rods *s*, *s*, within the control of the person in charge of the boiler, who ought to open it as soon as the action of the engine ceases, or even if at any time the steam-gauge S should indicate a pressure higher than the maximum, although no steam escapes by either of the safety-valves. The utmost vigilance and care should be exercised to keep these valves constantly clean and in good working condition.

To supply the waste from evaporation, and to maintain a uniform level of the water in the boiler, a self-acting feeding apparatus is employed, which, in the example now before us, is of a peculiarly perfect and complete description. In any convenient position upon the top of the boiler, the feed pipe M is erected; it is of considerable dimensions and of a height greater than that of a column of water which will counterbalance the ordinary pressure of the steam in the boiler. By means of the pipe N, the feed pipe is prolonged within the boiler until it is brought nearly to the bottom. It is surmounted by a small cistern O, which is kept constantly filled with water up to a certain level by means of the feed pump of the engine; the only communication between this cistern and the feed pipe is by a small conical hole in the bottom of the former, guarded by the plug or valve *e*, which is connected by a slender rod to the lever *d*. This lever carries at one end a counterweight *g*, and at the other a long rod passing through the small vertical pipe *f*, of the same length as the feed pipe, and terminated at the lower extremity in two curved arms upon which rests the stone float P. By this arrangement it is obvious that should the water in the boiler fall below its proper level, the float P descends, and, by its connection with the lever *d*, opens the small valve *e*, through which the water flows from the cistern into the feed pipe until the proper level be restored. The weight *g*, serves to counterbalance that part of the weight of the float which is not neutralized by its immersion in the water. The cisterns O, O are kept supplied with water by the pipe V communicating with the feed pump of the engine; flanges *m*, *m*, being cast upon the cisterns for the purpose of effecting the connection, while the surplus water that may be thrown up by the pump is led off by the waste pipe V′.

This feeding apparatus differs from that generally in use, in having the rod attached to the float passing through the boiler by means of a *water joint*, instead of a stuffing box; the superiority of this arrangement in securing tightness without friction is very obvious. Another important collateral advantage is gained by the use of the pipe *f*: in case the feed pump should, by any accident, cease to act, and the water consequently get low in the boiler, the steam would make its escape through the pipe *f*, before any serious injury could happen, indicating, the instant the water gets below the end of the pipe, that some derangement has occurred.

The draught of the furnace is regulated on a similar principle to that of the feeding apparatus. In the flue near the chimney is placed a damper or plate *i*, moving perpendicularly in a grooved frame in such a manner as entirely to stop the draught when shut. This damper is suspended by a rod and chain passing over two pulleys, the opposite end of the chain being attached to a hollow cast-iron float *h*, capable of moving freely within the feed pipe M; the float and damper are in equilibrium when the former is partially immersed in the water. The height of the column of water in the feed pipe varying according to the pressure of the steam within the boiler, it is obvious that, by the arrangement here described, any undue amount of pressure is speedily corrected by the descent of the damper, and *vice versa.*

On the front of each boiler is fitted an apparatus for indicating the height of the water within the boiler. It consists simply of a close glass tube Q, communicating at its upper extremity with the steam within the boiler, and at the lower with the water. The attendant in charge of the boilers is thus enabled at a glance to ascertain their condition; but as this apparatus is liable to frequent derangements and requires to be often cleaned, two stop-cocks *j, j*, are provided upon the communicating pipes, which admit of the glass tube being emptied and cleaned without stopping the action of any of the other parts, or losing any of the steam. Another indicator of the height of the water is also applied; it is called the self-acting water gauge, and is represented at R in our Plates. A small standard is fixed upon the top of the boiler, with a pulley over which a chain passes connected at one end with a copper wire and stone float *k*, which, by its rising or falling, indicates the exact variation that takes place in the height of the water, a graduated scale and index being provided for that purpose. The steam gauge S consists of a bent tube partially filled with mercury, one end of which communicates with the steam within the boiler, while the other is open to the air, and contains a small rod floating on the surface of the mercury; this floater, rising or falling with the mercury, points out, upon a graduated scale *l*, the degree of pressure of the steam in the boiler. T is the man-hole, with its door or cover, for giving admission to the interior of the boiler when it may be necessary to repair or clean it; and U the blow off cock at the bottom of the boiler for allowing the water to flow off.

Having thus described the general arrangements adopted in this set of boilers, we shall conclude by giving a brief notice of the various kinds of valves, and the modes of regulating their action.

The stop-valve, enclosed within the chest H, consists of a flat disc *n*, of gun metal, with a conical edge fitting steam-tight into a suitable seat formed of the same material, and retained in its proper place by a projecting tail, as shown in figs. 7 and 9. On its upper surface is jointed a brass spindle, which passes through a stuffing box on the cover of the chest, and is surmounted by a wrought-iron socket, through which passes a lever *q*, having its centre of motion in a small upright standard. So long as the valve is required to remain shut, the weight *r*, is kept suspended upon the long arm of this lever, but when it is to be opened it is transferred to the opposite extremity, as shewn by the dotted lines in fig. 9.

The safety-valve at J is of a very peculiar kind, and appears to be admirably adapted to obviate the risk of accidents in boilers of every description. The valve *p*, itself is spherical, being a hollow brass ball filled with lead, of such weight as to press with as many pounds per square inch as the maximum pressure of steam in the boiler is intended to be. Being perfectly free from friction, the obvious effect will be that at the very instant the steam arrives at this degree of elastic force the ball will be raised, and a discharge will immediately take place. The waste steam may be withdrawn either at the side, (as shown in the drawings), or at the top of the chest J. To protect the ball from being injured or tampered with, and to prevent its being thrown off its seat, the cover is formed in the peculiar manner distinctly represented in figs. 7 and 10. Mr. Hick, the inventor of this valve, recommends that it should in no case be used as a substitute for, but only in addition to, the ordinary safety valve, which is that represented enclosed within the chest K, and which is placed within the control of the person in charge of the boilers, as has been before explained.

At fig. 16 is shown a species of safety valves usually placed directly upon the boiler, and which are very frequently employed in place of the spherical valve, especially for low pressure engines. The bearing surface is flat instead of conical, and it is loaded with a weight which shall balance the ordinary pressure; and when it is required to let the steam escape, another weight, sufficiently heavy to raise the valve, is suspended upon the end v', of the lever v. Fig. 17 shows another form of safety valve which is sometimes employed in the same conditions and for the same purposes as the preceding.

Literal References.

A, A, the shell, or outside plates of the boilers.
a, a, a, cross-stays for strengthening the boilers.
B, B, flue under the boiler.
C, cross flue at the further end of the boiler.
D, the first lateral flue.
E, the cross flue at the front of the boiler.
F, second lateral flue.
G, the chimney.
b, the furnace door.
c, c, doors for giving access to the flues for the purpose of cleaning them.
H, H, the stop valve chests.
I, I, the steam pipe.
J, inaccessible safety valve chest.
K, chest for the safety valve, under control of the engine keeper.
L, pipe for conveying away the waste steam from the safety valve chests.
M, M, the feed pipes.
N, continuation of the feed pipe within the boiler.
d, d, levers for self-acting feed geer.
e, the valve or conical plug for do.
f, f, upright pipe, forming a water joint for the rod attached to
P, the stone float within the boiler.
g, g, counterbalance weights for the stone float.
h, hollow cast-iron float for self-acting damper regulator.
i, i, the dampers.
Q, glass tube and gauge cocks for indicating the level of the water.
j, j, pipes communicating between the boiler and the gauge cocks.
R, the self acting water gauge.
k, stone float for do.
S, the steam gauge for indicating the amount of pressure.
l, graduated scale of steam gauge.
T, the man-hole frame.
U, the blow-off cock.
n, n, the stop valves.
o, the safety valve enclosed within the chest K.
p, inaccessible spherical safety valve.
q, r, lever and ball for regulating the stop valves.
s,s, lever and rods for bringing the safety valve *o*, within the control of the engine keeper.
t, t, the furnace bars.
u, u, bearers or supports for the fire grate.

PLATES XXVI., XXVII.

DEAL SAWING MACHINE, PATENTED BY J. M'DOWALL, ENGINEER, JOHNSTONE, RENFREWSHIRE.

This machine differs from the common saw-mill frame in having the motive power included within itself, which involves also a modification in the arrangement of several of the working parts. On this system, each sawing frame is converted into a separate engine, perfectly distinct from the adjoining ones, and capable of being worked or stopped at the pleasure of the attendant, without affecting any other of the frames contiguous to it. It admits also of any frame being driven faster or slower than the others, as the qualities or natures of the various kinds of timber under operation may require.

The machinery may be constructed without the steam cylinder and piston, and driven by an ordinary stationary steam engine, there being a fast and loose pulley mounted upon the main shaft, upon which a belt from the actuating part of the stationary engine may be passed.

Fig. 1 shows a front elevation of the machine, looking to the face of the saws.

Fig. 2 is a side elevation, as seen at the left of fig. 1.

Fig. 3 is a section taken vertically through fig. 2.

General Description.

The standards A, A, or main upright framings of the machine, are bolted to the foundation, and otherwise firmly secured in their places by flanges; they are also bolted to the main beams that support the flooring B, B, thus dividing the machinery between the upper and lower apartments; the one of which contains the engine or driving power, while the other is occupied by the saw-frame machinery strictly so called.

The cylinder of the steam engine, by the agency of which the sawing machinery is driven, is shewn at C; it is supplied with steam in the usual manner, through slide valves contained in the valve casing *a*, which receives the steam from the boiler through the main steam pipe D, and the induction pipe E. Having served its purpose in the cylinder, the steam is discharged by the eduction pipe F into the waste-steam pipe G.

The main driving shaft H, H is mounted upon bearings fixed upon the top of the standards A, A. This shaft carries two fly-wheels I, I; two eccentrics J and K, and also two crank plates L, L. Into these last are fixed the crank pins *b*, *b*, which are connected by the rods *c*, *c*, to the cross-head M, carrying the frame to which the saws *d*, *d* are attached. This frame travels in the parallel slides or guide pieces *e*, *e*, bolted to the cheeks of the framing A, A. It will be seen that while the eccentric J, is working the slide valve which admits and discharges the steam to and from the cylinder C, the eccentric K is imparting motion to the spur wheels *f*, *f*, for the purpose of feeding or leading forward the timber; this is effected by means of the rod *g*, and paul or click *h*, which drives the ratchet wheel *i*, and pinion f', these being keyed upon the same carrying stud.

These spur wheels *f*, *f*, which are of rather a fine pitch, give a gradual progressive motion to the timber under operation, by means of the system of meter wheels and shafts *j*, j'; and it will be perceived that of these wheels the four lower pairs are so mounted as to communicate to the shafts carrying the four fluted iron feeding rollers *k*, *k*, *k*, *k*, an equal progressive motion simultaneously and in the same direction, which thus feeds in the timber shown at T, T. By these means, the timber in its progress through the machine, is gradually advanced and presented to the teeth of the saws, as they are driven up and down by the reciprocating action of the piston rod *l*.

It will also be understood that the extent of progressive motion given to the feeding rollers through the agency of the system of geering just described, may be varied and adjusted by an arrangement of mechanism which admits of the paul *h*, in its receding movements slipping over a greater or less number of teeth of the ratchet wheel *i*, and thus producing a quicker or slower advance of the timber, according to the nature of its grain or the number of saws which are acting upon it.

The governor by which the supply of steam to the valve box is regulated, is placed at the top of the machine, and is of the ordinary construction; the working parts of the governor being attached to the circular plate or disc of metal *n*, which is driven by the friction of the periphery of the revolving collar *o*, keyed upon the main shaft. This mode of driving the governor simply by friction is preferred, because it admits of ready adjustment by sliding the collar nearer to, or farther from, the centre of the plate *n*, thereby regulating the speed of the governor, as may be desired by the attendant, whenever it is necessary to vary the speed of the engine to suit the different qualities of the timber to be operated upon.

Action of the Machine.—Having now described the general construction of this sawing machine, it remains to show the mode of putting it into operation. It is to be understood that the engine for communicating the power is upon the high-pressure principle, and that the engine shown in the drawings is supposed to represent one of that construction, of about seven horse power, which will be generally found sufficient for ordinary timber. Let it also be observed that this engine is shown in the position of half-stroke.

The engine being first started in the ordinary way, by opening the throttle valve, which is placed in the induction pipe E, and thus admitting the steam, its expansive force will cause the piston *l*, to rise, and by means of the cross-head *m*, and side rods m' m', will draw up the shaft M, carrying the frame of saws *d*, *d*. The rods *c*, *c*, connecting this bar to the pins *b*, *b*, will drive round the crank plates L, L, and thereby communicate rotatory motion to the main shaft H, which will impart its impetus to the fly-wheels I, I, and, at the same time, cause the eccentrics J and K to revolve; the eccentric J giving motion to the slide valve in the usual manner, through the rod *p*, the levers *r*, *r*, and shaft *q*; and thus the necessary reciprocating action will be continuously imparted to the frames containing the system of saws *d*, *d*. These saws are of course adjustable in their respective frames, so that any number may be employed, and at any required distance apart.

It has been before stated that, when the engine is in operation, the other eccentric K will cause the timber to advance to the teeth of the saws as required, by the system of geering which works the four fluted feeding rollers *k*, *k*, *k*, *k*. It will also be perceived by reference to fig. 1, that each of the vertical shafts *j*, *j*, has a feather or raised key upon its surface; these are for the purpose of carrying the two wheels *j'*, *j'*, round with them, and at the same time allowing these wheels to slide upon their respective shafts while in motion, in case of any inequalities upon the surface of the timber, causing the upper feeding rollers *k*, *k*, to rise or fall. To keep the wheels in geer, the boss of the pair of horizontal wheels is elongated, and has a groove turned upon it in order to embrace the end of the shaft of the feeding rollers, as shown in fig. 1. Ordinary feeding tables with delivering rollers are placed in front and behind the sawing frame, to conduct the deals and timber to be cut on to the plate N, N, upon which they are supported while under the operation of cutting; and upon this plate, at the feeding end or front of the machine, there is placed a central guide piece, or *fence s*, against which the deals are to be pressed by the springs *t*, *t*, having friction rollers at their ends for that purpose.

These springs are adjusted to the required tension by the small handle and screw *u*, *u*. The tables placed behind the machines to receive the timber after it has been cut into boards are, in other respects, similar to the front tables. It will be seen that there are two weights W, W suspended from the shafts of the upper pair of feeding rollers *k*, *k*, by the rods *v*, *v*, in order to keep them down upon the top edge of the timber with sufficient pressure to resist the upward lift of the saws when cutting. These weights are also suspended at the centre by a small chain and pulley from the shaft *w*, at one end of which is a ratchet wheel *x*, provided with a large boss having mortices cut in it for the purpose of receiving a small crow-bar, in order to lift the weight and ease the top rollers when required.

Literal References.

A, A, A, A, the standards or upright framing of the machine.
B, B, the beams and flooring of the mill.
C, the steam cylinder.
a, the steam valve casing.
D, the main steam pipe.
E, the induction or branch steam pipe.
F, eduction pipe from the engine.
G, the waste steam pipe.
H, the driving-shaft of the engine
I, I, fly-wheels upon do.
J, eccentric for working the steam valve.
K, eccentric for working the feed motion of the saws.
L, L, crank plates of the engine.
b, *b*, the crank pins in do.
c, *c*, connecting rods between the crank pins and
M, the great cross bar of the saw frame.
d, *d*, the saw-plates.
e, *e*, guides of the saw frame.
f, *f*, *h*, *i*, ratchet and spur wheels for the feeding motion.
g, rod for conveying the motion of the eccentric K to the feeding geer.
j, *j*, *j'*, small bevil wheels and shafts for conveying the action of the feeding geer to
k, *k*, *k*, *k*, the fluted feeding rollers.
N, N, the shears, or table for supporting
T, T, the log of timber undergoing the action of the machine.
l, the piston rod of the engine.
m, the cross-head of do.
m', *m'*, side-rods connecting the cross-head with the saw frame.
n, *o*, two circular plates or discs, which, by their friction, drive the governor.
p, eccentric rod of the valve geer.
q, *r*, traverse shaft and lever of do.
s, a guide piece for regulating the motion of the timber to be cut.
t, *t*, springs with rollers acting on the deals during their passage through the machine.
u, *u*, screw and handle for adjusting the tension of the springs.
v, *v*, rods for transmitting a pressure to the top feeding rollers from
W, W, two weights suspended at the bottom of the machine.
w, *x*, shaft and ratchet wheel for relieving the strain of the weights W, W.

PLATES XXVIII., XXIX.

VENEER SAWING MACHINE, BY JAMES GRAY AND CO., GLASGOW.

These Plates represent the most approved modern form of the veneer cutting saw, with its appendages. The drawings are executed with the utmost care and accuracy from a machine at present in operation in Glasgow, which performs its functions in the most successful and satisfactory manner.

The object of this machine is to cut up logs of mahogany, rosewood and other fine woods, into thin slices or leaves for the purposes of veneering. From ten to sixteen of these laminæ may be formed from a thickness of one inch of wood; and when it is considered that the logs usually vary from 9 to 36 inches in breadth, and that it is absolutely essential that the slices should possess a smooth and uniform surface, some idea may be formed of the nicety of the operation, and of the rigidity and steadiness of action requisite in the different parts of the machine, in order to the efficient performance of its work.

Enumeration of the Figures.

Fig. 1 is a front elevation in which the great circular saw is denoted by the shaded part of the figure.

Fig. 2 is a vertical transverse section of the machine, showing the construction of the travelling frame work, to which the log to be cut is attached, and the mode of communicating to it its longitudinal and transverse motions.

Fig. 3 is a general plan of the machine, in which the great circular saw is supposed to be sectioned in order to show its construction. Here the geering for giving motion to the travelling frame is most distinctly exhibited.

Fig. 4 shews the form of the centre of the great saw frame as seen from the back. The small figure underneath is a section of one of the radiating arms.

Figs. 5, 6 and 7 are details, upon an enlarged scale, of some of the more important parts of the machine which could not be fully understood from the general views.

General Description.

The principal feature of this machine is the great circular saw, the construction and relative dimensions of which are most clearly exhibited at figs. 1 and 3. The saw itself is composed of a great number of segments *a*, *a*, *a*, secured by countersunk screws to the rim of a circular cast-iron frame A, A, A, which has previously been reduced to a perfectly plane surface by turning. This frame consists of a series of arms radiating from a central eye, and terminating in a broad flat ring, projecting slightly in front of the arms, and upon which the segmental saw-plates are disposed and fixed in the manner stated above. The arms are perfectly flat upon the front edges, but towards the back they are formed with feathers or flanges, tapering gradually from the circumference to the centre, in order to give them the requisite degree of strength and rigidity. The spaces between the arms both in front and at the back of the saw frame are filled up by plates of thin sheet-iron, giving to the whole the appearance of a segment of a great sphere. It is accurately hung upon a strong horizontal cast-iron shaft B, B, provided with fast and loose driving pulleys C, C, by means of which a rapid motion is communicated to it by a belt from the crank shaft of a steam engine or other prime mover. The shaft B revolves in bearings *b*, *b*, secured to suitable sole-plates set upon walls of solid masonry D, D, for the purpose of raising the saw to the proper elevation above the floor of the mill.

While the axis of the saw frame remains stationary during the operation of the machine, the log of wood which is to be cut up is urged forward against the saw by a gradual progressive motion, which, by the arrangements adopted in this machine, is rendered self-acting, and may be regulated to any required velocity. For this purpose the log is firmly secured to a sliding carriage E, E, E, framed together of cast-iron beams, of such form and dimensions as to give the requisite degree of rigidity without unnecessary weight. This frame is situated in front of the saw and is adapted to slide transversely upon a lower carriage F, F, which is supported at the points *c*, *c*, *c*, upon the longitudinal guide rails *d*, *d*, *d*, *d*, firmly bolted to the upper surface of two parallel timber beams G, G. The guide rails *d*, *d*, are accurately planed and set truly parallel to each other and to the face of the saw, and the supporting pieces *c*, *c*, are formed of brass for the sake of diminishing the friction. A long rack H, H, fixed to the under side of the carriage F, F, and geering with a spur wheel I, upon the horizontal shaft J, serves to give the requisite longitudinal feeding motion to the carriage which supports the log to be cut up; this motion being derived primarily from the shaft B of the saw, through the intervention of a system of self-acting geering to be presently described.

That part of the frame E, E, which is contiguous to the great saw, presents a broad vertical plane surface, of dimensions corresponding with those of the largest logs which may at any time be subjected to the operation of the machine. Upon this face is fixed a timber frame *e*, *e*, *e*, consisting of longitudinal and diagonal spars, upon the latter of which the log to be cut up is immediately sustained. The surface of the diagonal spars is formed truly plane and parallel with the face of the great saw and with the guide rails *d*, *d*, and the log adheres to them by means of a film of glue interposed. The spars are set forward at an angle of about 45° in order to oppose a greater surface of resistance to the downward strain of the saw during the process of cutting.

It has been already remarked that the frame E, E is susceptible of a transverse motion upon the lower carriage F, F; this is for the purpose of moving the log in the direction of its thickness in setting it for a fresh cut. For this end the lower cross rails of the frame E are provided with dove-tail sliding pieces *f*, *f*, (see fig. 7), accurately fitted to work in corresponding bearings attached to the lower carriage. The position of the upper in relation to the lower carriage is determined by the screws *g*, *g*, which revolve in fixed bearings bolted to the latter, and pass through brass nuts secured to the former in the manner indicated in fig. 6. On each of these screws is fixed a small worm-wheel; these work into endless screws formed on the longitudinal axis *h*, *h*, to which motion is imparted by means of the handle *i*. By this arrangement it is obvious that, on turning the handle *i*, the frame E, with its appendages, will be advanced towards or withdrawn from the face of the saw, according to the direction in which it is turned, and each of the screws *g*, *g*, being situated towards the extremities of the frame, and both being acted upon simultaneously, the log which is being cut up will retain its parallelism to the saw in whatever position it may be set in relation to it.

The self-acting motion for working the sliding carriage longitudinally, and thereby feeding the saw, is derived from the main shaft B, B, or axis of the latter. A small pulley *j*, is fixed upon this shaft towards its front extremity, and communicates motion by means of a belt and pulley *k*, to a short intermediate axis working in bearings fixed to a cast-iron frame K, K, bolted upon the front wall D. This short shaft carries a set of cone-pulleys *l*, which by means of an open belt, conveys motion to a corresponding set *m*, upon an axis situated at the level of the driving shaft J of the rack H. These last named shafts are connected by a train of spur geering for the purpose of reducing the velocity, which may readily be modified to a considerable extent, according to the nature of the wood acted upon, by means of the cone-pulleys *l*, and *m*. The bearings of the various shafts in this train of wheel work, as well as of the driving shaft J itself, are supported upon a rectangular cast-iron frame L, bolted firmly to the masonry forming the floor of the mill. Upon the axis of the cone-pulley *m*, is fixed a pinion *n*, geering with a wheel *o*, whose shaft is susceptible of an endlong motion for the purpose of disengaging the wheel and pinion which it carries from the rest of the geering when required. The pinion *p*, works into the wheel *q*, on the third motion shaft which transfers its motion to the main driving shaft J by means of the pinion *r*, and wheel *s*, which complete the train. By this system of geering the velocity of the great shaft B, which is very considerable, (about 65 revolutions per minute), is reduced to a very slow and steady progressive motion of the frame to which the log to be cut up is attached.

When it is required to reverse the motion of the sliding frame E for the purpose of bringing the log back to the commencement of a fresh cut, the second motion shaft is thrown out of geer by means of the handle *u*, connected with it by a rod *t*, bringing it conveniently within the control of the attendant in charge of the machine. A pulley *v*, on the end of the intermediate shaft of the feed geering, communicates motion by a cross belt to the pulley *w*, turning loose upon the end of the third motion shaft. A coupling *x*, however, is provided upon this shaft, by means of which the pulley *w*, may be engaged with it, and thereby communicate a rapid motion to the driving shaft J, in the contrary direction to the feeding motion. This is effected by simply moving the pulley *w*, endlong upon its axis, by means of the handle *z*, connected with it by the rod *y*, when its arms coming into contact with the projecting pins of the coupling *x*, communicate its motion to the shaft, and continue it so long as the pulley is retained in that position. This arrangement will be best understood by reference to fig. 2.

Action of the Machine.—The log having been fixed to the spars *e*, *e*, in the manner stated above, and its position in relation to the saw having been adjusted by means of the longitudinal and transverse motions, according to the thickness of the veneer, the machine is set in action by shifting the driving belt from the loose to the fast pulley C. A rapid rotatory motion is thus communicated to the saw, and at the same time the log is urged slowly and steadily towards it, in obedience to the self-acting feeding motion described above. The veneer, as it is separated from the log, glides off the back of the saw-frame A, A, and when the saw has penetrated the entire log, falls off and is removed. The self-acting motion is then reversed, by sliding the wheel *o*, and pinion *p*, out of geer by means of the handle *u*, and engaging the pulley *w*, with the third motion shaft by the help of the handle *z*. The log having been thus returned into its original position in front of the saw, the pulley *w* is disengaged, and the log again adjusted for cutting off the next slice, by turning the handle *i*. The feeding geer is then again thrown into action by means of the handle *u*, and the process is repeated as already described. The surface velocity of the saw in the example now before us is about 3063 feet per minute.

Literal References.

A, A, the saw frame.
a, *a*, *a*, the segments of the saw.
B, B, the shaft on which the saw frame is hung.
b, *b*, plummer-blocks of the shaft B.
C, C, fast and loose strap pulleys hung on the shaft B.
D, D, walls of masonry for supporting the plummer-blocks *b*,*b*.
E, E, cast-iron frame to which the log to be cut is secured.
F, F, lower frame fitted with longitudinal and transverse slides.
c, *c*, brass pieces fixed to the carriage E, and resting upon
d, *d*, *d*, *d*, the longitudinal guide rails.
G, G, parallel beams of timber for supporting the rails *d*, *d*.
H, H, toothed rack fixed to the frame F.
I, J, spur wheel and shaft giving motion to the rack H.
e, *e*, *e*, sparred timber frame for fixing the log to be cut up.
f, *f*, dove-tail slides for transverse motion of the frame E.
g, *g*, transverse screws for setting the frame E.
h, longitudinal axis carrying endless screws for working the guide-screws *g*, *g*.
i, crank-handle for working the shaft *h*.
j, pulley on main shaft for giving motion to the feed geer.
k, driving pulley on intermediate shaft.
K, K, bracket for supporting do. do.
l, cone pulleys on do. do.
m, cone pulleys on first motion shaft of the feed geer.
L, sole-plate for the bearings of the various shafts of the feed geer.
n, *o*, *p*, *q*, *r*, *s*, spur wheels and pinions constituting the feed geer.
t, *u*, rod and handle for disengaging the forward feed motion.
v, *w*, pulleys for communicating a backward motion to the feed geer.
x, *y*, *z*, clutch, coupling and handle for throwing the pulley *w* into or out of geer.

PLATE XXX.

SINGLE SCREWING MACHINE. BY WREN AND BENNETT, MANCHESTER.

This machine is distinguished for simplicity of construction, and lightness of appearance, and for a peculiarly ingenious and elegant combination of mechanism for regulating the position of the cutting dies during their action upon the bolt on which the thread is to be formed.

Fig. 1 is a side elevation, and fig. 2 a sectional plan of the machine. Fig. 3 is an end view, shewing the frame of the die-holder, and fig. 4 a corresponding end elevation with the die-holder and outer supports removed, in order to shew the mechanism of the chuck or revolving head for fixing the bolt to be screwed.

General Description.

The framing consists of two cast-iron cheeks A, A, bolted by flanges to the floor of the work-shop, and retained at their proper distance by two stay-bolts *a*, *a*, and by two stronger wrought-iron rods B, B, similarly disposed, and which, being prolonged to a considerable distance in front of the machine, serve also as guide rods for the gland frame carrying the cutting dies, or the nut to be screwed; the outer extremities of these rods are supported by two upright rods *b*, *b*, also secured to the floor by bolting. The lower rails of the fixed framing A, A have flanges formed interiorly, for supporting the bearings of the shaft carrying the driving cone-pulley C, of six speeds, commanded by a similar pulley, upon an intermediate or *counter* shaft, to which motion is imparted from the main shaft by fast and loose pulleys in the usual manner. At the back of the machine the cone-pulley shaft is prolonged beyond its bearing, and carries the pinion *c*, of 13 teeth, geering with a large spur wheel D, of 112 teeth, fixed upon the end of the main or screwing spindle E, which is of considerable diameter and made hollow, in order at once to ensure the necessary rigidity without undue weight of material, and to admit of the introduction of bolts, when required, into the interior of the spindle. This spindle is fitted to revolve very smoothly and steadily in brass bushes fitted into bearings provided for their reception in the framing A, A.

In front of the machine a strong cast-iron chuck, or revolving frame, is keyed to the end of the screwing spindle, and is fitted with the necessary appliances for adjusting and fixing the bolt which is to undergo the operation of screwing. These consist of a square socket *a*, capable of sliding in an oblong recess in the chuck F, and provided with a pinching screw *e*, for fixing the bolt in the interior of the socket, which is itself retained in its position, when properly adjusted, by the lateral pinching screws *f*, *f*, working through the rim of the chuck. The adjustment of the socket *d*, consists simply in shifting its position in the recess into which it is fitted, until the axis of the bolt coincides with that of the spindle; the action of the machine itself, when set in motion, will, in most cases, cause the socket to assume its proper position.

The die-holder G is fitted to slide freely in a longitudinal direction upon the guide rods B, B, in obedience to the action of the revolving bolt upon the cutting dies which it carries. It consists of an open cast-iron frame, within which a wrought-iron piece H is accurately fitted to slide transversely upon V-shaped guides inserted into the frame, and working in the correspondingly formed edges of the piece H, which encloses the cutting dies *j, j;* these are likewise formed with angular edges to retain them in their places, and to admit of their proper adjustment.

In order to cut the thread of a screw with accuracy in a machine such as the present, it is of course essentially requisite that the centre of the cutting dies should coincide with the axis of the bolt to be screwed, and as these must be capable of approaching or receding from each other according as the thread is more or less completely formed, it becomes a problem of some difficulty to accomplish this without displacing the common centre. The very ingenious manner in which this is effected in the machine now before us constitutes, as we have before remarked, its principal peculiarity. A double screw *h*, works at once through the frame G and the transverse sliding piece H, and its point bears against the back of one of the dies *j, j*, the position of the other in relation to the frame H being fixed and invariable. That portion of the screw which works through the frame G, (being fitted to a brass nut riveted into its edge), is formed with a *right-hand* thread, and with a pitch only half of that part which passes through the interior frame H, and which, on the contrary, is a *left-handed* screw. By this arrangement it will be remarked that, if the common centre of the dies, in any given position in relation to each other, be once properly adjusted into the axis of the machine, that centre will be maintained in all other positions, and whatever may be the amount of space between them. Thus supposing the dies, as represented in fig. 3, to be in the axis of the machine, when brought close together, in the proper relative position for forming the *full thread*, then, the pitch of the inner end of the screw *h*, being one-fourth of an inch, and that of the outer end one-eighth of an inch, it is obvious that one revolution of the screw will cause the frame H and right hand die to move one-eighth of an inch to the right; while at the same time the other die will be left free to move one-fourth of an inch to the left, and thus the common centre of both will be preserved, at the same time that the dies are expanded for the reception of a bolt on which the thread is to be formed from the commencement. A hand wheel *i*, is affixed to the outer end of the screw *h*, for the purpose of regulating its motions, and a small index is attached to it in order to point out when the thread is full, &c.

The mode in which this machine acts in screwing bolts is so obvious as to require no further explanation. When nuts are to be screwed, they are fixed in suitable sockets in the frame G, while the tap is fixed into the screwing spindle in the same manner as we have already described in reference to the bolt.

PLATES XXXI., XXXII.

DOUBLE SCREWING MACHINE, BY WILLIAM MORE, GLASGOW.

This machine, which is one of the most powerful and complete of the class to which it belongs, exhibits, both in the arrangement of its geering and in many of the subordinate details, a marked difference to that which forms the subject of the preceding Plate. It is capable of cutting the threads of screws of 4½ inches diameter, and, unlike most other machines of the kind, the geering is so adjusted, that both sides of the machine can be employed simultaneously upon bolts and nuts of different sizes.

Enumeration of the Figures.

Fig. 1 is a general side elevation, and fig. 2 a plan of the whole machine.

Fig. 3 is an end elevation, and fig. 4 a vertical transverse section, showing very distinctly the general arrangement of the geering.

Fig. 5 is a longitudinal section of the main screwing spindle with its attached geering and die-holder, the latter being sectioned upon the line *A—B*, fig. 6.

Fig. 6. is an enlarged front view of the die-holder, with the external covering plate removed; and fig. 7 represents the form of the sliding cross-head for retaining the bolt during the operation of screwing.

General Description.

The two main standards A, A are fixed upon a strong cast-iron sole-plate B, which extends the whole length of the machine, and is securely bolted to a stone foundation. Upon these standards all the geering of the machine is mounted. The driving spindle J is placed intermediate to the screwing spindles E and N, and carries the three-speed cone I, by which motion is communicated to the machine. The spindle J has a bearing in each of the two standards, and carries the fixed pinions *a*, and *b;* also the wheel U and pinion *d*, which are cast together, but run loose on the spindle. The pinion *a*, geers into the two spur wheels S and F, which are loose upon the two screwing spindles E and N; and the pinion *b* geers with the spur wheel K, which is loose upon a spindle L immediately under the driving spindle J. The wheel U geers into the wheel R, and the pinion *d*, into the wheel O, which is fast on the screwing spindle N. The spindle L has its bearings also in the two end standards, and carries, besides the wheel K, another wheel M geering with the loose wheel G on the screwing spindle E; also a fixed pinion *c*, which geers with the wheel P keyed upon the screwing spindle N. The wheels K and M are loose on the shaft L, but are both fast upon a common hollow boss, so that motion being communicated to the wheel K, the wheel M will be carried round in the same direction and with an equal velocity.

The arrangement of the wheels on the large screwing spindle N, is fully shown at fig. 5. This spindle is provided with a hollow boss Q, on which are fitted the wheels R and S; the former being fixed upon the boss Q, while the latter is loose, but may be engaged with it when required, by means of the sliding clutch or coupling box T, which can be worked from either end of the machine by the double handle Z, Z. The wheels O and P are fast upon the spindle N, as before stated.

The smaller screwing spindle E carries only the two loose clutch-wheels G and F, either of which can be brought into action by the sliding coupling H, which is worked by the double handle Y, Y. This handle is placed upon a small rocking shaft *l*, carried on two brackets *m*, *m*, bolted to the sole-plate of the machine, and has the clutch fork *n*, keyed upon it, so that in moving the handle from the vertical position, the clutch will be brought into geer with one of the wheels G, F, on the spindle E. The handle Z, Z is in like manner fixed upon a cross shaft *o*, carried by the brackets *p*, *p*, similarly fixed upon the sole plate; but this shaft, besides the clutch fork *q*, for working the clutch T, on the hollow boss Q of the screwing spindle N, has a second fork *q′*, for working a clutch on the spindle L, to engage and disengage the loose boss of the wheels K and M. But these two clutches are so fixed in relation to each other that one of them only can be in action at once; consequently, when the wheel S is engaged by the clutch T, the wheels K and M will be loose on the shaft L, and *vice versa.*

The screwing spindles are of malleable iron to ensure strength, and are made hollow to allow the bolts to pass into them as they are screwed. The chucks are fast upon the ends of the spindles, and to these the die-holders are bolted. The die-holder of the smaller spindle is of the common form, and fits into a dove-tail recess V, from which it can be removed and have its cutters changed at pleasure; but that for the larger spindle is of a somewhat more complicated construction, as will be observed from the face view of it given in fig. 6. It consists of a strong plate or disc W, annularly recessed to receive a ring X, flush with its exterior surface. This ring has a portion of its circumference cut into teeth, which geer with a worm fitted to turn on an axis *h*, in the plate W, and which can be worked by means of a handle or key. Thus according as the worm is turned in one direction or the other, the ring X will receive a corresponding motion, which will change the position of the dies or cutters *f*, *f*, *f*, in relation to the axis of the chuck. For this purpose three spiral recesses *g*, *g*, *g*, are formed on the inner circumference of the ring, into which the exterior ends of the dies project, abutting against their outer edges. The dies are accurately fitted into radial slots prepared for their reception in the ring *e*, which forms part of the disc W, and the whole is covered by the thin plate *j*. In this Plate are also three narrow radial slots *k*, *k*, *k*, (fig. 3), through which pass three small round pins projecting from the dies, for the purpose of guiding them radially. From this construction it is obvious that, if the ring be made to pass through a small arc from right to left, the dies will be forced to approach the centre; and conversely, if the motion be from left to right, they will be allowed to expand and receive a bolt of larger diameter.

In the operation of screwing, the head of the bolt is caught in the cross-head D, (fig. 7), which fits loosely between the guide rods C, C, along which it slides towards the chuck as the thread is progressively formed, while the screwed part passes within the screwing spindle. When nuts are to be tapped, they are inserted into similar cross-heads fitting the guide-rods C, C, at the opposite end of the machine, and the taps are turned by the square holes in the ends of the spindles.

Action of the Machine.—The arrangement of the geering above detailed being kept in view, the action of the machine will easily be understood. Thus, supposing motion to be communicated to the speed cone I, if the clutches T and H be in geer with the wheels S and F respectively, these wheels will be driven by the pinion *a*, with a speed proportioned to their respective diameters, and in opposite directions. Meantime, the clutch on the under shaft L being out of action, the wheels K and M will be loose upon it, and the shaft itself will be made to revolve idly by means of the wheel P, which geers with the pinion *c*, upon it. The motion of the wheel F will be directly communicated to the screwing spindle E and its chuck V; but that of the wheel S will be transferred to the hollow boss Q, and thence to the wheel R, which geers with that marked U. But this last being loose upon the driving shaft, and fast with the pinion *d*, will transfer its motion through the wheel O, to the screwing spindle N, and so communicate a reduced speed in the ratio of the numbers $\frac{a \times R \times d}{S \times U \times O}$. But let the clutch T be disengaged, the clutch H remaining in geer as before; then the under clutch will engage the wheels K and M to their shaft L, and consequently this shaft will be driven by the pinion *b*, which geers with the wheel K; thus driving the wheel P and the screwing spindle N in the opposite direction to its former motion, and with a speed determined by the ratio of the numbers $\frac{b \times c}{K \times P}$. Again, let the clutch H be brought out of geer with the wheel F, and engaged with the wheel G, then the spindle E will receive an increased speed in the opposite direction to its former motion. Thus the two screwing spindles may be driven in either direction independently of each other, and may be employed at the same time to screw bolts and nuts of different sizes and pitches of thread.

PLATES XXXIII., XXXIV.

VERTICAL DRILLING MACHINE, BY JOSEPH WHITWORTH AND CO., MANCHESTER.

The machine represented by these Plates is one of the most elegant and perfect of its kind which has yet been produced, and is distinguished by that high degree of completeness, durability, and mechanical excellence which characterize the tools manufactured by Messrs. Whitworth. It is provided with all the mechanism requisite for adapting machines of its class to every variety of circumstances in which they may be required to act; but its most remarkable feature consists in the very peculiar and ingenious mode in which the feed motion of the tool is produced and rendered self-regulating.

The Plates contain both general and detailed views of the machine, which is represented in such a manner as to make the construction and action of its various parts perfectly intelligible.

Enumeration of the Figures.

Fig. 1 is a sectional elevation of the machine in the plane of the axis of the driving shaft and work table.

Fig. 2 is an external front elevation of the whole machine.

Fig. 3 is a sectional plan, shewing the construction of the upper surface of the work table, and the mode in which it is guided in its vertical motion, whilst being adjusted to the required height. Fig. 4 is a sectional elevation of the same parts as are represented in fig. 3.

Figs. 5 and 6. An elevation and plan of the mechanism by which the rotatory and rectilinear motions of the work table are produced.

Figs. 7 and 8. An elevation and plan of the friction collars employed in obtaining the self-acting feed motion of the drill spindle.

Figs. 9, 10, 11 and 12. Detached views of the various spindles and wheels for the adjusting motions.

General Description.

The various working parts of this machine are sustained upon a strong cast-iron framing or pillar, indicated by the letters A, A, A; it is formed of a single casting, and is fixed by three strong bolts upon the sole-plate T, T, which is intended to rest evenly and firmly, though without being bolted, upon the floor of the work shop. On the upper portion of the main frame a strong bracket is cast, which serves to carry the outer ends of the driving spindle *b*, and back-speed spindle of the machine. Upon the spindle *b*, is the driving cone B of three speeds, the spur wheel C and the bevil pinion D. The cone-pulley B is fitted to revolve loose upon its axis, and only communicates motion to it by means of the spur wheel C, which is keyed upon the spindle, and to which the cone *can be attached* by a stud-pin and nut at *c*. This wheel geers with the pinion E on the same spindle which carries the wheel M; this in turn geers with the pinion N, which is fast upon the end of the cone B, and consequently runs loose upon the spindle *b*.

This arrangement is in every respect the same as the ordinary back-speed of a lathe, and serves the same purpose. Supposing the back-speed removed, the cone being driven by its belt causes the spindle *b*, to revolve, in consequence of its attachment to the fixed wheel C, and thus motion is conveyed directly to the bevil wheel D on the end of the spindle. This again geers with a similar bevil wheel F, on the drill spindle G, G, which is free to slide vertically in the eye of the wheel, while at the same time it is prevented, by means of a sunk feather, from revolving independently of it. In this way three different degrees of quick speed may be communicated to the drill. But let the back-speed be in geer, as represented in the drawing, (fig. 1), and let the stud-pin *c*, be removed, and the cone thereby loosened from its attachment with the wheel C; the motion now communicated to it will not drive the shaft *b*, directly as before, but the pinion N being fast upon it will give motion to the wheel M, upon the same spindle with the pinion E. This last will therefore make the same number of revolutions as M, but being smaller in diameter, will convey a proportionally reduced velocity to the wheel C with which it geers. Now the wheel C being fast on the shaft *b*, conveys the same diminished speed through it to the bevil pinion D, and this again to the drill spindle G, G. This reduced velocity may, of course, be varied as before by placing the belt on a larger or smaller pulley of the driving cone B.

Behind the pinion E there is a recess cast in the framing, to allow it to enter when the back-speed wheels are to be thrown out of geer, which is done by simply sliding their spindle longitudinally towards the back of the machine; and it may be remarked that the back-speed is only required to be in action when the machine is employed in boring holes of upwards of an inch and half in diameter.

The wheel F is cast with a long hollow boss, which is turned and fitted to revolve in brass bushes, fixed into the lower branch of the carrying bracket in the usual way. The drill spindle G, G passes, in the manner already described, through the boss of the wheel F, which thus serves to guide its lower extremity; the upper end works in a collar similarly fitted into the upper branch of the bracket at *a*. By this arrangement the drill spindle is guided vertically in its ascent and descent, throughout the entire range of its travel; in the drawings it is shewn at its lowest limit.

To the top of the drill-spindle the back-balance weight H is attached by a jointed lever and guide link, which embraces the top of the spindle and moves upon a vertical guide-rod, firmly bolted to the upper surface of the standard A, A. The object of this arrangement is to cause the drill-spindle to rise when the feeding apparatus is relaxed.

The drill-spindle is screwed towards the middle of its length; it is there embraced by two screw-wheels J, J between which it turns, and which serves the purpose of a nut to feed down the spindle in the operation of drilling. This is effected by an arrangement which will presently be described.

The article which has to undergo the operation of boring is set upon a broad cast-iron table K, to which it can be firmly held down and adjusted by T-headed bolts and glands in the usual way, when thought necessary. The table, it will be observed, is recessed and grooved to receive and retain the heads of the holding bolts, as fully shewn in the horizontal section fig. 3, and the side view fig. 4. When the article is much smaller in area than the surface of the table, the head of the fixing bolt can be inserted at any convenient point of the surface by the recesses *k*, *k*, and slid backward or forward until it passes below the projecting ledges of its recess, where it is then retained. The common mode of running two sets of inverted dove-tail grooves at right angles to each other, lengthways and across the table, seems to us, however, preferable to the arrangement here described.

The table is itself supported upon the sole of the large carriage bracket L, which is strengthened by two strong ribs cast on its under side. This bracket is attached to the framing A, A by two pieces Y, Y, which are bolted upon it and planed true to the angle of the inverted bevil edges of the broad face-rib of the main frame. These edges are also planed where they meet the oblique faces of the pieces Y, Y. By this means a joint is formed which allows of a sliding motion vertically, but does not admit of any deviation laterally. When the slides become loose by wearing of their surfaces, the pieces Y, Y admit of being tightened by means of pinching screws fitted against them through continuous projections cast on the bracket. This arrangement is fully explained by figs. 3 and 4.

This bracket is raised and depressed by means of a hand-crank applied at *u*. Upon this spindle is fixed the spur pinion U, (shewn in combination in figs. 1 and 2, and separately in figs. 11 and 12). This pinion geers with a spur wheel V on the same spindle with the pinion W, which geers in turn with the rack *w*, fixed immoveably into the frame A, A of the machine. It is thus evident that on turning the hand-crank on the shaft *u*, the motion will be transmitted to the bracket, which will either be raised or lowered, according as the crank is turned in one direction or the other.

The table K has a double movement upon the sole of the carriage bracket, one being circular and the other in the direction of the length of the table. The circular movement is effected by means of the hand-crank *p*, upon the spindle *y*, which is carried in bearings formed on the box X. On the spindle *y*, is formed a worm, which geering with the worm wheel P, on a hollow stud projecting downwards from the table K, conveys the motion of the handle *p*, to the table. The construction of these parts is fully represented in the enlarged detail views, figs. 4, 5 and 6.

The lateral movement of the table is effected by a different arrangement. An oblong recess is cast in the sole of the carriage bracket L, with projecting ribs *e*, *e*, on its under side, to serve the double purpose of giving strength to the sole-plate, and of forming guides against which the cover of the travelling box X may slide, the surfaces in contact being planed true for that purpose. The motion is communicated by means of the handle *q*, upon the short spindle which carries the bevil wheel Q, (shewn separately in fig. 10), geering with a similar wheel R upon the end of the guide screw *d*, which has its bearings *r*, *r*, in the sole-plate of the carriage bracket, and works through a long nut formed in the cover of the travelling box X. It is thus obvious that on turning the handle *q*, the carriage and the feed table to which it is attached will be carried along in the direction of the length of the screw.

By means of the two motions above explained, any point of the table K can be brought under the axis of the drill; and by means of the vertical motion it can be placed at any convenient height.

The feed of the tool during the operation of boring is obtained, as before stated, by means of the two screw-wheels J, J, by an arrangement of parts which forms the chief peculiarity of this machine. On the axes of these wheels are placed two concentric discs or pulleys, the peripheries of which are embraced by the friction collars S, S, (shewn separately at figs. 7 and 8). The bearings of the axis being attached to the framing A, A of the machine, it is obvious that when it is in motion, provided the discs be prevented from revolving, the wheels J, J will likewise remain at rest; but, the screwed part of the drill spindle revolving between them, they will act as a fixed nut, and cause the spindle to descend through a space equal to one thread of its screw during every revolution. Again, suppose the discs and wheels free, the screw of the spindle, instead of descending, will simply cause the wheels J, J to revolve on their axes through a space equal to one tooth during every revolution of the screw. Between these extremes any amount of feed or downward motion of the drill spindle may be obtained by simply retarding the motion of the wheels by means of the friction collars S, S, which embrace the small discs on their axes; for if the friction of the collars be less than to prevent entirely the motion of the wheels, and at the same time greater than to allow a tooth to pass during a revolution of the spindle, a downward motion of the latter, proportional to the amount of retardation of the discs, will thus be produced. It will be observed that the friction collars are formed in halves, with bosses joining the segments, pair and pair; these bosses are screwed internally in opposite directions, the thread of the one to the right, and that of the other to the left; and being placed on a rod *s*, correspondingly screwed right and left, they are made to approach or recede from each other according as this rod is turned to the right or left. The rod *s*, is prolonged downwards, and has a handle fixed on its lower end, by which the required degree of friction can be adjusted at pleasure. The collars S, S are lined with copper, for the purpose of increasing the friction; for it is manifestly of advantage to obtain as much friction as possible on the discs with the least possible amount of pressure.

This arrangement has also the advantage of allowing the tool to be speedily withdrawn; for on relaxing the friction collars the balance weight H will raise the spindle in ordinary cases; and should it be prevented by any cause from rising, the balance weight can be assisted in its office by a hand-wheel placed on the axis of one of the wheels J; the wheel and screw being thus for the time converted into a pinion and rack.

Literal References.

A, A, A, the upright framing of the machine.
a, the upper bearing of the drill spindle.
B, the driving cone pulley.
b, the driving spindle.
C, spur wheel on the driving spindle.
c, catch for engaging the spur wheel C with the cone pulley.
E, spur pinion on the back motion shaft.
D, F, a pair of meter wheels for conveying the motion of the driving shaft to
G, the drill spindle.
H, back balance weight of the drill spindle.
J, J, screw wheels acting on the drill spindle.
K, the work table.
k, recesses in do. for inserting bolt heads.
L, a bracket for supporting the work table.
M, N, second spur wheel and pinion of the back motion.
O, hollow axis and collar connecting the work table with the mechanism of its various motions.

d, d, guide screw for the longitudinal adjusting motion.
e, e, rectangular recess on the bracket L for guiding the longitudinal motion of the work table.
P, worm wheel for the rotatory motion of the work table.
p, y, handle and worm shaft for giving motion to the wheel P.
X, the carriage containing bearings for the various parts of the adjusting motions.
Q, R, a pair of small meter wheels for giving motion to the guide screw *d.*
q, handle for working these wheels.
r, r, bearings of the guide screw *d.*
S, S, the friction collars for the downward feed motion of the drill spindle.
s, s, a screwed rod and handle for regulating the friction of the collars S, S.
T, T, sole-plate of the machine.
U, V, W, spur wheels and pinions of the mechanism for setting the feed table to the required height.
u, small shaft for giving motion to this geer by means of a handle.
w, w, vertical rack fixed to the standard A, for raising the feed table bracket L.
Y, Y, adjustible pieces forming a guide for the vertical motion of the bracket L.

PLATES XXXV., XXXVI.

PATENT SELF-ACTING SLOTTING MACHINE, BY JOSEPH WHITWORTH AND CO., MANCHESTER.

The machine represented in these Plates affords an excellent illustration of the perfection to which modern toolmaking has arrived, and which the distinguished firm whose name it bears has done so much to advance. In it we discover an admirable combination of the requisite qualifications of strength, simplicity, steadiness of action, and perfect adaptation to the various circumstances under which machines of the class to which it belongs are destined to act. The example now before us is of very considerable dimensions, and the degree of rigidity absolutely essential in the framing of machines of this description can only be attained by forming them of a proportionate weight of metal; yet so judicious is the disposition of the material that the expert mechanic, whose eye has been educated to judge of beauty in machines by their adaptation to the ends proposed, will, at a glance, pronounce the design of the present specimen to be faultless.

The characteristic feature which distinguishes this Slotting Machine from others of its class consists in the peculiar mechanism employed for giving motion to the slotting bar. The usual mode of converting the rotatory motion of the driving shaft into the required vertical rectilinear motion, is by the intervention of a crank and connecting rod; an exemplification of which arrangement will be found in Plates LXIV., LXV. This method is sufficiently simple and obvious, but it has the disadvantage of impressing a constantly varying velocity upon the slotting bar; so that the tool cannot be made to act upon the work throughout all parts of the stroke at the rate of speed at which it will cut most advantageously. In the machine now before us, this inconvenience is obviated, and great additional steadiness and regularity of action are obtained, by employing a screw to give motion to the slotting bar. This screw may be formed with one, two, or more threads, according to the rate of speed at which the machine is usually required to travel; this velocity being in most cases very low as compared with that of the driving shaft, leads us to notice another important advantage attending the use of the screw: namely, that by affording the means of directly reducing the speed to the required amount, all intermediate wheel work is dispensed with, and the driving geering rendered exceedingly simple.

To transfer the motion of the screw to the slotting bar it is of course necessary to make use of a nut, or of some other contrivance which shall serve the same purpose. The advantages above enumerated, the value of which will be sufficiently obvious to the practical mechanic, could not have been made available in practice, had not some efficient substitute for the nut been devised, seeing that they would have been more than counterbalanced by the loss of power arising from the effects of friction and torsion between the nut and a screw working within it at a high velocity, and in opposition to a considerable strain. This substitute has been contrived and most successfully applied by Messrs. Whitworth. The principle of this ingenious contrivance, the details of which will be fully described below, consists in reducing to the utmost the number of points at which the power of the screw is transferred to the slotting bar, and in interposing between these points and the slotting bar an arrangement of friction discs or rollers. By this means the power is brought into effective operation in a very direct manner, and the friction is reduced to a minimum.

The method of converting a rapid rotatory into a slow rectilinear motion, by means of a screw shaft and friction discs, has also been applied by Messrs. Whitworth to their planing machines and self-acting spinning mules.

Enumeration of the Figures.

Fig. 1 is a side, and fig. 2 a front elevation of the machine. In the first of these views a portion of the framing, and in the second, of the slotting bar, is removed in order to shew the screw shaft and friction disc apparatus.

Fig. 3 is a transverse section of the slotting bar, and of the guides in which it works.

Fig. 4 is an external elevation, fig. 5 a vertical section, and fig. 6 a plan of part of the self-acting feeding mechanism detached. By the arrangements delineated in these figures, the machine may be made to communicate either a reciprocating or a continuous rotatory motion to the shaft which drives the various feeding screws, according to the nature of the motion which it may at any time be required to impart to the table on which the work is set.

General Description.

This machine is independent, and contains within itself all the shafts, spindles, bearings, &c., which are necessary for giving motion to its various parts. The framing A, A, A consists of a massive cast-iron sole, capable of being bolted to a stone foundation, two strong uprights, and two cross stays; these several parts being all firmly bolted and keyed together, so as to form one rigid and inflexible mass. In the centre of the cross stays, and on the front of the machine, are situated the dove-tail guides *a, a, a, a,* in which the great vertical slide or slotting bar B, B is fitted to work accurately and smoothly; the construction of the guides, as shewn in fig. 3, is such as to allow of a certain amount of adjustment for the purpose of correcting the tendency of the bar to relax in its bearings from the effects of wear and tear. At the lower extremity of the slotting bar a recess is provided for the reception of the tool-box C, which is secured in its place by numerous pinching screws.

Parallel with, and immediately behind the centre of the slotting bar, the driving screw-shaft D is fitted to revolve in bearings situated in the centre of the cross stays of the framing. On the upper end of the screw shaft is fixed a bevil wheel *b*, geering with two small pinions *c*, and *d*, the former of which is keyed fast to the driving shaft E, revolving in suitable bearings on the summit of the machine, and set in motion by the strap pulley F, while the latter is cast upon the end of the prolonged boss of the pulley G, which is accurately bored out and fitted to revolve smoothly upon the shaft E, but with a motion altogether independent of it.* By these arrangements it is obvious, that if the driving belt, which runs continuously in *the same* direction, be made to work alternately upon the pulleys F and G, it will impress upon the screw shaft D a rapid rotatory motion alternately in *contrary* directions, according as the belt commands the pulley which is in connection with the pinion to the right or to the left of the centre of the screw-shaft, and this alternating motion of the screw-shaft, being transferred to the slotting bar B, produces the requisite reciprocating motion of the cutting tool. The intermediate pulley H runs loose upon the driving shaft, and serves to carry the belt while it is being shifted between the others, and when the machine is not in action. The shifting of the belt is effected by means of the sliding strap-fork *e*, whose motions are regulated according to the required length of the stroke by the self-acting geering to be presently described.

The anti-friction apparatus for replacing the nut in the transference of the motion of the screw shaft to the slotting bar, consists of a pair of discs *f*, *f*, mounted upon axes which are carefully fitted to revolve in adjustable bearings *g*, *g* situated in strong brackets I, I, projecting behind the slotting bar. The friction discs are set face to face upon each side of the screw, and are formed with projecting rims adapted to enter into the spaces between the threads which, by this arrangement, are made to act against the strain only at two points diametrically opposite in each disc; these points being in the direct line of the strain, and the friction upon them being further reduced in the ratio of the diameter of the discs to that of their axes.

The motions of the slotting bar are regulated, and the length of its stroke determined by the two adjustable catches *h*, and *i*, fitted to slide up and down in a groove formed on one side of the slotting bar, and in which they may be secured to the required position by bolts passing through them. These catches are so formed and set as to come into contact, at the termination of the upward and downward motion of the bar, with two small friction rollers *j*, *j*, fitted into the arms of the bell-crank J, the lower catch working against the upper arm and *vice versa*. This action gives motion, alternately in opposite directions, to the cross shaft K, which is the axis of the bell-crank, and to the opposite extremity of which is fixed an arm *k*, formed into a short segment of a bevil wheel, and working into a similar segment *l*, fixed about the middle of the vertical shaft L. The alternating motion thus imparted to this latter axis is, by a similar arrangement of toothed segments *m*, *m*, at the upper end, transferred to the lever *n*, which being in direct communication with the sliding strap-fork *e*, shifts it alternately between the pulleys F and G at the termination of the required stroke, and thus reverses the motion of the slotting bar.

The work which is to be subjected to the action of this machine is laid upon, and firmly bolted to the circular table M, which is susceptible of three distinct self-acting feed motions, one along the bed of the machine, another across the bed at right angles to the former, while the third is a motion of rotation round its own axis. The two first of these motions are imparted to the table by means of suitable slides and feed screws in the usual manner. The lower slide N is fitted to travel longitudinally upon the upper surface of the bed of the machine, which is accurately planed and prepared for its reception; the feed screw which commands it is shewn in dotted lines in fig. 2. The upper slide O is in a similar manner adapted to travel at right angles to the direction of the former, and is set in motion by the screw shewn in dotted lines in fig. 1. The circular motion of the table is obtained by means of an endless screw and worm wheel P, working upon the upper surface of the slide O. By the arrangements adopted in this machine any of these motions may, in the simplest manner, be imparted to the table, in either direction and with any required velocity. They have their common origin in the alternating rotatory motion of the vertical shaft L, which, for this purpose, is prolonged downwards to the level of the upper surface of the bed plate, where it terminates in the peculiar combination of mechanism represented, on an enlarged scale, at figs. 4, 5 and 6, by means of which the workman in charge of the machine is enabled to control and regulate, or to disengage altogether, the self-acting feed motions, as circumstances may require.

This is effected by the following ingenious arrangement: —An arm *o*, having two adjustable stops or studs *p*, *p*, bolted to its lower surface, is fixed fast upon the shaft L by a small key, and consequently partakes of its reciprocating motion. Under it is situated another double arm or lever *q*, *loose* upon the shaft, and carrying at its outer extremity a spring paul or detent *r*, fitted to engage with the teeth of a ratchet wheel *s*, which has its centre of motion also in the shaft L, but is in no other respect connected with it. The adjustable stops *p*, *p*, of the arm *o*, coming into contact with the arm *q*, at each stroke of the machine, carry it round through an arc proportional to the distance at which they are set from each other, which may be varied according to the velocity at which it may be required to drive the feed screw. The paul *r* serves to give continuous rotatory motion to the ratchet *s*, which is keyed to the boss of a bevil wheel *t*, externally fitted to revolve in a bearing provided for its reception in the bracket Q, while internally this boss serves as the bearing for the lower end of the shaft L. The continuous motion of the wheel t, thus obtained, is by it communicated to the short transverse shaft R, by means of either of the two small pinions u, u', according to the direction in which the longitudinal feed screw is required to turn; these pinions being fitted to slide into or out of geer with the wheel for that purpose (see fig. 1). A pair of meter wheels v, v', serve to convey motion from the shaft R to the longitudinal feed screw, and complete this part of the self-acting mechanism. A handle S is provided at the opposite extremity of the lower feed screw, for the convenience of working it by hand, in which case the bevil wheel v, is slid forward upon its axis R our of geer with the wheel v'. The small handle shewn on the outer end of the arm q, is for the purpose of turning the paul r, out of contact with the teeth of the ratchet wheel s, when it is required to discontinue temporarily the self-acting motion of the longitudinal feed screw.

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* This arrangement of driving geer is fully explained and illustrated in our Article on Tools, Part III., page 193.

When the transverse or circular feeding motions of the table M are to be used, it is necessary to continue the alternating motion of the shaft L upon the short cross shaft R. For this purpose the paul r, is thrown out of contact with the ratchet wheel s, and the latter is simply attached to the arm q, by means of the screw bolt w. It is to be remarked, however, that in this, as in the former case, the distance between the stops p, p, regulates the amount of feed given at each stroke of the machine. The outer end of the cross shaft R carries a segment of a spur wheel x, geering with a sliding rack T, to which is attached a long rod U, U, working in bearings at the front of the machine. This rod is provided with an arm y, fitted to slide upon it to any required position, where it may be secured by a pinching screw. A small rod z, serves to connect this arm either with the lever a', or b', according as the circular or transverse feed is to be communicated to the table M. In our drawings the former is shewn in action; the reciprocating motion of the lever a', being converted into a continuous rotatory motion on the spur ratchet c', by means of the paul d', the form of which is such that it may be made to act upon the wheel c, in either direction. The motion thus generated in the wheel c', is communicated by it to the small spur wheel e', fast upon the axis of the endless screw f, which drives the worm wheel P. The transverse motion of the table is obtained by transferring the rod z, from the lever a', and connecting it with the lever b', on the end of the transverse feed screw.

Literal References.

A, A, A, the framing of the machine.
a, a, a, dove-tail guides of the slotting bar.
B, B, the slotting bar.
C, the tool box.
D, the vertical screw shaft for giving motion to the slotting bar.
b, c, d, bevil wheel and pinions for transferring to the screw shaft the motion of
E, the driving shaft of the machine.
F, G, H, fast and loose pulleys on the driving shaft for stopping and reversing the action of the machine.
e, the sliding strap fork for moving the driving belt along the pulleys.
f, f, friction discs for transferring the action of the screw shaft to the slotting bar.
g, g, the bearings of the friction discs.
I, I, brackets for the reception of the bearings g, g.
h, i, adjustable catches on the slotting bar for working the reversing and feed motions.
J, j, j, bell crank with friction rollers, acted on by the adjustable catches
K, intermediate shaft of the reversing and feed motions.
k, l, toothed arms for transferring the motion of the horizontal shaft K to
L, L, the vertical shaft of the reversing and feed motions.
m, m, toothed arms for transferring the motion of the vertical shaft to
n, a lever attached to the sliding strap fork e.
M, the work table of the machine.
N, lower slide for the longitudinal motion of the feed table.
O, the upper slide for the transverse motion.
P, worm wheel for the circular motion of the work table.
o, p, p, arm with adjustable catches, for originating the feed motions.
q, r, s, lever with paul and ratchet, for converting the reciprocating motion of the shaft L into a continuous rotatory motion.
Q, lower bearing of the vertical shaft L.
t, u, u', bevil wheel and pinions for transferring the feed motion to the cross shaft R.
v, v', a pair of meter wheels for giving motion to the longitudinal feed screw.
S, handle for working the longitudinal feed screw by hand.
w, a screw bolt for fixing the feeding geer so as to give motion to the transverse and circular feed screws.
x, a toothed segment working into
T, U, a horizontal rack and rod, carrying
y, an adjustable stud, connected by a slender rod z, with the levers and wheels of the circular and transverse motions, marked a', b', c', d' and e'.

PLATES XXXVII., XXXVIII.

HAND PLANING MACHINE.

The vast advantages, in point of economy, accuracy and celerity, which have been found to result from the employment of the Planing Machine in our workshops, were considered, for some time after its introduction, to be exclusively confined to the execution of works of considerable magnitude. More recently, however, it has been discovered that its uses were not so limited; but that it could be advantageously employed in the dressing of articles of the smallest dimensions, such as keys, straps, &c.; and even in situations where no other motive power could conveniently be applied but the hand of the workman, its superior efficiency in every respect over the tedious and imperfect mode of dressing previously in use, has led to its general adoption in such circumstances. In the Plates now before us, we have given a specimen of a small planing machine of the kind last alluded to, in which the table for carrying the work is set in motion by the hand of the workman.

Fig. 1 is an end elevation of the machine as seen from the front.

Fig. 2 is a side elevation upon a somewhat smaller scale; and

Fig. 3, a ground plan corresponding to fig. 2.

The machine is raised to a convenient elevation above the floor of the workshop by the two cast-iron frames A, A, supporting the extremities of the bed frame B, B, which consists simply of a species of open cast-iron box, formed with two parallel triangular grooves on its upper surface, and traversing its entire length. Towards the back of the machine is situated the upright frame L, L, for carrying the tool-box and its slides; the face of this frame is accurately dressed to a plane vertical surface, against which the transverse slide C is fitted to work up and down, in order to admit of the tool-box being set to the height of the article which is to be planed. This is effected by means of the two vertical screws *f*, *f*, suspended from fixed bearings in the top of the upright frame, and working through the projecting pieces *a*, *a*, cast on the back of the slide C; these screws are simultaneously set in motion by an arrangement of bevil wheels V, V, worked by a crank-handle upon the axis W, as distinctly shewn in fig. 1, and by this means the horizontal slide is constrained to move parallel to itself in whatever position it may be required to be set. When properly adjusted, it is secured to its place by four bolts passing through upright slots formed in the face of the fixed framing L, L.

Upon the face of the slide C, a rectangular piece D is fitted to traverse its entire length, being provided with adjustable dove-tail slides for the purpose of retaining it upon its place, and preventing any lateral movement. A screw *b*, *b*, which may be turned either by hand or by a self-acting apparatus to be presently explained, serves to give motion to the piece D; being fitted to revolve in bearings in the ends of the slide C, and working in a nut fixed on the back of the carriage D. On the anterior surface of this last is carried the vertical slide *c*, *c*, the piece *d*, *d*, being interposed, which admits of a small amount of circular motion on the stud bolts *h*, *h*, passing through circular slots in the disc *d*, and provided with pinching nuts to retain it in the position desired. By this construction the tool slide may be set to any required angle, as in other planing machines. The tool-box moves vertically between the cheeks of the cutter-slide *c*, its position in relation to the article to be planed, or in other words, the depth of the cut, being regulated by the screw *k*, which is surmounted by a small hand-wheel. The tool-box consists of two parts *e*, and *g*; the piece *g*, which is that to which the tool is fixed, being susceptible of a small amount of rotatory motion upon the vertical slide *e*, for the purpose of enabling the tool to pass over any obstructions on the return stroke of the travelling table.

The travelling table E, E, on which the article to be subjected to the action of the machine is placed, consists of an oblong rectangular cast-iron plate, having on its under surface two parallel ribs of a triangular section, which are accurately planed and fitted into the corresponding grooves on the bed frame B. The work to be planed is fixed to its place by bolts sliding in dove-tail grooves provided upon the upper surface of the table, and motion is imparted to it by means of a pinion F, geering with a rack M fixed to the table, and turned by a hand-cross H, upon the outer extremity of the horizontal shaft G. This mode of applying the motive power to the machine is considered preferable to the simple crank-handle, inasmuch as it admits of the workman standing constantly in an erect position, thereby enabling him to exert his strength more advantageously, and at the same time to have his eye constantly fixed upon the work before him.

The self-acting transverse motion, or feed, of the tool slide is obtained from the motion of the travelling table E, by means of a small tappet or catch *s*, which is fitted to slide upon a dove-tail rib cast upon the edge of the table, and to which it may be fixed by a pinching screw in any required position, according to the length of the travel of the table. The tappet *s*, is so adjusted as to come into contact, at the end of the return stroke, with a short lever working upon a stud, and having at the other end of its axis another lever *n*, and counter-balance weight *o*; the lever *n*, is traversed by a dove-tail slot in which is placed an adjustable stud, connected by a slender rod *r*, with a similar lever *m*, which works loose upon the end of the feed screw *b*, *b*. The lever *m*, is provided with a paul *l*, so placed as to engage with the teeth of a ratchet wheel K fixed upon the feed screw. By this arrangement it is obvious that when the tappet *s*, comes into contact with the short lever against which it works, it will cause the paul *l*, to slide over a certain number of the teeth of the ratchet wheel K, and that on being released from this contact, the weight of the counter-balance *o*, will bring the levers *m*, and *n*, to their original positions, and so turn the feed screw *b*. These levers are slotted in order that the studs carrying the connecting rod *r*, may be set at any required distance from their respective centres of motion, and consequently, the number of teeth over which the paul is carried depending upon the positions of the studs, the amount of feed motion of the screw *b*, *b*, can be regulated at pleasure.

PLATES XXXIX., XL.

SELF-ACTING PLANING MACHINE, BY ARCHIBALD MYLNE, GLASGOW.

The preceding Plates exhibit the planing machine in its simplest form, depending for all its motions upon the immediate superintendence of the workman, and limited to the execution of works of small dimensions and of the most elementary forms. The machine represented in the Plates now before us is of a much more complicated description, and indeed possesses some peculiarities which render it worthy of a place among the higher order of tools of its class. It is self-acting in all its motions, and is especially adapted to facilitate the production of machines of a similar description with itself.

Fig. 1 is a side elevation, and fig. 2 an end view of the entire machine as seen from the front. Fig. 3 is a longitudinal section of the slides which regulate the motions of the cutting tool, shewn in combination, and figs. 4, 5, 6, 7 and 8 shew the separate parts in detail. Fig. 9 exhibits the mode of setting the tool slides so as to operate upon oblique surfaces.

General Description.

The bed-frame *k*, *k*, on which the table for carrying the work traverses, consists of a single casting, and is supported upon the legs or standards I, I. Its upper edges are formed into two projecting angular rails, which are planed true and fitted to corresponding V-shaped grooves on the under surface of the travelling table *m*; an arrangement, as will be observed, the inverse of that adopted in the preceding example, and indeed generally in planing machines. The work is fixed upon the travelling table *m*, by bolts with dove-tail heads, which slide in longitudinal grooves of corresponding forms. At the back of the machine is situated the upright frame F, F, for carrying the tool-slides; it is formed of two side brackets, bolted down to strong flanges cast on the bed-frame *k*, and joined together at the top by a cross piece, which gives to the frame the necessary rigidity and strength. The entire face of the frame thus constructed is planed true and polished; this is requisite in respect of the upright faces, as upon these the vertical slide *a*, *a*, moves when it is elevated or depressed by the screws *u*, *u*, to adapt it to the height of the work; and the cross piece is polished to avoid unseemly contrast of appearance.

Motion is imparted to the machine by means of the driving belts A and B, the former being an open belt working over a pulley of considerable dimensions, for the purpose of communicating a slow and powerful motion to the table during the forward or operative part of its stroke; the latter is a cross belt driving a smaller pulley for the return motion, which is thus made considerably quicker than the forward motion. By means of the strap guide C, these belts may be shifted either by hand or by the action of the machine itself from fast to loose pulleys, as may be observed at fig. 2, in which they are represented upon their loose pulleys. These pulleys revolve on a common spindle or shaft, working in suitable bearings in the frame G, G, which is bolted at one end to the side of the bed-frame, while the opposite end is supported by small columns secured to the masonry of the foundation. The strap-guide C passes through the bed-frame, and is provided with a handle at its outer end, for the purpose of regulating its position by hand when necessary; the opposite end is flat and works through an upright socket attached to the frame G, to prevent it from turning round, while it admits of its sliding longitudinally.

On the outer extremity of the pulley shaft is keyed a small spur pinion D, geering with a wheel E, fixed to a strong shaft passing through the bed-frame of the machine, and carrying two small pinions which give motion to the sliding table by working into the racks *l*, *l*, upon the under side of it, as shewn in the end elevation. These racks and their respective pinions are set so as to have the teeth of the one opposite to the space of the other, thus rendering the motion more smooth and uniform than if only a single rack were employed.

The travelling table *m*, as has been before remarked, will move in either direction according as the driving geering is commanded by the open or cross belts. The self-acting mechanism for effecting the requisite alternating motion of the strap-guide C is as follows: On the outer edge of the table *m*, are placed two adjustable stops or tappets *n*, *n'*, which slide in dove-tail grooves, and may be fixed at any required distance from each other, according to the length of the object which is to be planed, and to its position on the table. These tappets are so formed as to come in contact alternately, as the table is moved backwards and forwards, with the arms of a double lever *g*, which is fixed to a boss or hollow shaft working freely upon the driving shaft; the back tappet *n'*, stands further out than the other, so that each can act upon only one arm of the lever *g*. Upon the same hollow shaft which forms the centre of motion of the double lever, a short lever *j*, is fixed; this is jointed by means of a connecting rod *x*, passing longitudinally under the table *m*, to a similar lever fixed upon a horizontal shaft at the back of the machine, (see fig. 1). To this shaft are attached the double lever *p*, *p*, (the arms of which are at right angles to each other, and carry the adjustable weights *r*, *r'*), the levor *z*, for conveying motion to the reversing geer, and that marked *t*, for actuating the tool slides. The lever *z*, terminates in a screw, for the purpose of adjustment to the required amount of travel, and the nut to which it is fitted is connected by a short link to a lever which conveys the motion in a transverse direction to a longitudinal shaft M, working in suitable bearings under the bed-frame of the machine. The opposite extremity of this shaft carries a lever J, which is connected to the strap-guide C in such a manner as to cause the alternating angular motion of the shaft M to shift the driving belts A and B alternately from the fast to the loose pulleys and *vice versa*.

This mechanism, though somewhat complicated, is found to produce an exceedingly efficient and serviceable motion for the purpose for which it is intended. The mode of action is as follows: The machine is set in motion by pushing the belt guide C towards its further side, and thus the forward driving belt A is shifted on to the narrow pulley, which is the driver. The sliding table *m*, is thus put in motion, and travels slowly towards the back of the machine, until the tappet *n*, coming into contact with one of the arms of the lever *g*, turns it over, and, by means of the connecting rod *x*, imparts motion simultaneously to the levers *p*, *z*, and *t*. The upper weight *r*, is thus thrown off the balance, and, its gravitating tendency coming in aid of the force by which this is effected, the belt-guide C is pulled forcibly by the lever J, until the belt A is shifted to the loose pulley, and at the same time the cross belt B to its fast pulley, which produces the return motion of the table. When this has been continued to the required distance, the tappet *n'*, strikes its appropriate arm of the lever *g*, and turns it over in the opposite direction, thereby bringing the reversing geer into the proper position for effecting a fresh stroke, when the action of the machine proceeds as before, and may be continued as long as may be required.

The lever *t*, which is fixed, as before remarked, upon the same axis with those marked *p*, *p*, *z*, imparts the self-acting motion to the slides on the frame F, for causing the cutting tool to traverse over the surface of the work. A section of these slides is shewn at fig. 3, and detached parts at figs. 4 to 8, the same letters being used to denote the same parts in all the figures. We proceed to the enumeration of these parts.

Upon the face of the fixed frame F, the vertical transverse slide *a*, *a*, is fitted to bear very accurately and steadily; it is adjustable to the height required for the work under operation by means of two screws *u*, *u*, working through nuts fixed on the back of the slide *a*, and which may be simultaneously set in motion by the system of bevil wheels surmounting the machine, and commanded by the handle *o*. When set to the proper height, the slide *a*, is secured to its place by bolts passing through the upright grooves in the cheeks of the frame F. The cross slide *b*, which is of a square form, is fitted to move across the breadth of the machine, upon the face of the vertical slide *a*.

Motion is communicated to it by a screw which has its bearings *s*, *s*, in the ends of the vertical slide, and which passes through a nut attached to the back of the slide *b*, as shown in fig. 3. The slide is guided, and also securely retained, on its place by dove-tail edges formed on the back, and which fit accurately to corresponding dove-tails formed on the slide *a*. The upper dove-tail is made adjustable to allow for any wear of the surfaces which may take place. On the face of the slide *b*, is a graduated arc, to regulate the setting of the slide carriage *c*, which is attached to it by fixing bolts and nuts. The bolts have dove-tail heads, fitted to slide in an annular groove, (see fig. 4), and passing through two holes in the circular pieces cast on the edges of the carriage *c*. The carriage thus admits of being set and fixed at any required angle with the slide *b*, as shown by fig. 9; the details of the mode in which this is effected is explained by figs. 4 and 5, and partially by figs. 2 and 3.

In the carriage *c*, is a screw with bearings at its two extremities, formed in the metal of the carriage; it is retained in its place by a collar on its upper end, fitted into a recess over which passes a ring or washer, fixed to the carriage by pins screwed into the metal. On the lower end of this screw is fixed a small bevil pinion, which geers with one of a pair bolted firmly back to back, so that motion cannot be communicated to the one without the other being moved simultaneously in the same direction. The second of these pinions geers with a similar one upon the rod *q*, *q*, (fig. 2), and their common bearing consists of the angular groove formed by the junction of the backs of the wheels; this groove is accurately turned to fit the V-shaped edge of an annular recess cut in the centre of the cross slide *b*, (see figs. 3 and 4). The purpose of this arrangement of wheels shall be explained presently.

On the face of the carriage *c*, a projecting piece is cast, which is planed with dove-tail edges, (fig. 5), to receive the corresponding dove-tails of the slide *d*, (fig. 6). On the back of this slide a nut is fixed, through which the screw in the carriage *c*, passes, for the purpose of raising and depressing the slide, especially when set obliquely, according to the circumstances of the work under operation.

Upon the face of the slide *d*, is a second carriage *e*, in all respects similar to that marked *c*, except that it is smaller. It is attached to the slide *d*, also by bolts with dove-tail heads which work in an annular groove in the face of the slide *d;* the ends of the bolts passing through holes in the projecting pieces cast on the edges of the carriage, which can thus be set at any required angle on the face of the slide *d*.

The slide *f*, is similar to that marked *d*, and is attached in the same manner to the face of the carriage *e*; it is moved to any position required, within the range of its travel, by the screw of its carriage, with which it is connected by a nut, (figs. 3 and 8). The tool-box is attached to this slide by a joint or hinge, the construction of which will be easily understood from inspection of the above named figures. The use of the joint is to prevent the tool *h*, from cutting or rubbing upon the work during the return of the table; a very slight amount of motion of the joint is sufficient to allow the tool to clear the work. The mode of fixing the tool is by T-headed bolts and glands, as shown in figs. 2 and 3; in this last the tool and glands are removed, but the fixing bolts are dotted in their positions.

The mechanism of the slides being thus fully detailed, it remains to point out the mode in which the self-acting motion of the reversing geer is communicated to them. A small spur-wheel *y*, is fitted to revolve freely upon a stud bolted to a projecting piece cast on the end of the vertical slide *a*. Behind the wheel *y*, and upon the same centre of motion, a long slender lever *w*, works freely; it is provided with a spring catch or paul, which may be made to engage with the teeth of the wheel *y*, either above or below the lever *w*, which is connected with the lever *t*, already brought under notice, by means of the rod *v*. The spring catch being in geer, the wheel *y*, receives motion from the lever *t*, and being engaged with a pinion on the end of the screw *s*, *s*, it communicates its motion to the cross slide *b*, and its appendages. The amount of feed is adjusted by shifting the studs in the slots of the levers *t*, and *w;* and the direction in which it is desired to move the slide is regulated according as the spring catch is geered above or below the axis of the lever *w*.

This self-acting feeding motion is also communicated to the down-cutting slide *d*, by shifting the pinion from the end of the screw *s*, *s*, to the end of the small shaft *q*, *q*, (fig. 2). On this shaft is the small bevil pinion before alluded to in reference to fig. 3, and which is carried round by having a key projecting into a groove or slot formed continuously throughout the whole length of the shaft. This pinion is carried along the shaft by a projecting piece on the slide *b*, (fig. 4); and its motion is communicated to that on the end of the slide screw in *c*, (fig. 3), by means of the two similar intermediate wheels placed in the slide *b*, as above described.

The front slide *e*, is not usually attached to planing machines, but it is valuable where work is to be executed which requires two or three different angular surfaces to be planed; this can be accomplished in this machine by arranging the slides before commencing the work, no subsequent alteration being required. Fig. 9 shows the slides set at different angles.

Literal References.

A, driving belt for the forward or cutting motion.
B, cross belt for the return motion.
C, the strap-guide.
D, spur pinion on the driving shaft.
E, spur wheel on the main shaft.
F, upright frame for carrying the tool-slides.
G, horizontal frame of the driving geer.
a, *a*, the carriage of the transverse slide.
b, *b*, the transverse slide.
c, *c*, the carriage of the main vertical slide.
d, *d*, the main vertical slide, also capable of being set at any angle.
e, *e*, the carriage of the front slide.
f, *f*, the front slide, being also the tool holder.
h, the cutting tool.
g, hollow boss or shaft with levers for working the reversing geer, &c.

j, n, n′, levers on the boss *g*.
k, the bed-frame of the machine.
I, I, legs or brackets for supporting the bed-frame.
l, l, racks for working
m, m, the travelling table of the machine.
o, crank handle for raising or depressing the slides by the screws *u, u*.
p, p, r, r, x, z, levers rods and weights forming parts of the reversing geer.
J, M, lever and shaft actuating the strap guide C.
q, q, a slender shaft in the slide *a*, for working the down cutting feed.
s, s, guide screw for working the transverse slide *b*.
t, v, w, slotted levers and connecting rod for conveying motion to the slide screws.
y, a small spur wheel, acting as a ratchet, for converting the alternating motion of the lever *w*, into a continuous rotatory motion.

PLATE XLI.

BACK GEER TURNING LATHE, BY JOHN HETHERINGTON, MANCHESTER.

The drawings in this Plate exhibit very fully the mechanism of a common back-speed turning lathe, of considerable dimensions, and of the most approved modern construction. It is intended to be set upon the ordinary timber bed-frame, or *shears*, of the turning shop, and may be used either with or without the *slide-rest*.

Fig. 1 is a front elevation of the fixed head-stock, with its mountings; fig. 2, an end view taken from the back, and fig. 3, a plan of the same. Figs. 4, 5 and 6 are an end view, side elevation and plan of the shifting head-stock.

The *fixed head-stock* A, A is a strong cast-iron framing, formed with two uprights for carrying the ends of the running spindle, and a broad flat sole for giving the requisite stability. It is fixed firmly to its place by a T-headed bolt at each extremity; these are fitted into corresponding recesses cast in the head-stock, and are secured beneath the shears by nuts and washers. The *running spindle s*, which is steeled at both ends, is very carefully fitted into its bearings in the head-stock; and, to oppose the endlong strain of the lathe, it is supported by a steel pin *d*, called the *back-centre*, fitted to slide longitudinally in a bracket cast on the back of the head-stock, and adjustable by a jam-nut and pinching screw. Its opposite extremity carries the *chuck* or *face-plate c*, on which the object to be turned is bolted, provided it is of such a form as to admit of that mode of fixing; if it be of considerable length, it is carried upon steel *centres*, inserted into the ends of both running and *shifting* spindles. The driving cone pulley *b*, is fitted to turn loose upon the spindle *s*, as also the pinion *p*, which is firmly attached to the cone, and geers with a spur-wheel *h*, upon the *back shaft e;* this shaft also carries a pinion *i*, working into the spur-wheel *w*, fixed upon the running spindle. This train of wheels is called the *back speed*, and is only employed when great power is required, the shaft *e*, being capable of sliding backwards in its bearings in the brackets B, B, until the wheels which it carries are out of geer with those on the running spindle, which in that case is made to revolve at the same speed with the cone, by directly connecting the wheel *w*, with it by means of a pin or latch. When the back speed is in geer the shaft *e*, is retained in its place by pins passing downwards through the brackets B, B, behind its bearings; these pins may be removed at pleasure, to allow the shaft to slide backwards.

The *shifting head-stock* is of a somewhat similar form to that already described, but instead of carrying a revolving spindle it is fitted with a cylindrical stem *o*, (fig. 5), adapted to slide longitudinally in a socket accurately bored out for its reception, at exactly the same height as the bearings of the running spindle. A steel *centre* is inserted into one end of the spindle *o*, and forms the point of support for the object which is to be turned; the mechanism by which it is brought into contact with it is as follows:—At the further extremity of the shifting spindle *o*, is fixed a gland or cross-head *l*, connecting it with a cylindrical guide rod *m*, adapted to move freely through the end of the head-stock parallel to the spindle *o;* in the cross-head, between these, a suitable bearing is provided for the end of the screw *t*, which also works in the metal of the head-stock as in a nut, or, which is better, in a brass nut inserted and fixed firmly into the head-stock; the screw is turned by means of a hand-wheel *n*. By this arrangement the position of the spindle *o*, may be regulated with the utmost accuracy, so as to enable the workman, after having fixed the shifting head near the required place, to suspend the object to be placed in the lathe, upon the centre points of the opposite spindles, or to remove it at pleasure. In order to relieve the screw *t*, from the strain, and to prevent it from relaxing, a wrought-iron ring *u*, bored to the exact diameter of the spindle *o*, is inserted into a recess in its socket, and is tightened up by a nut made into the form of a handle *r*, and fitted to a screwed shank projecting upwards from the ring.

PLATE XLII.

NUT CUTTING MACHINE, BY ARCHIBALD MYLNE, GLASGOW.

This description of tool is convenient, and indeed indispensible in factories where the chief business is the construction of the more highly finished quality of machinery, in which the nuts are usually dressed to correspond with the other parts of the work. Such machines are not commonly employed by millwrights, although their use would often effect a material saving of time in the fitting shop, and especially in out-door work, in reducing the nuts, and consequently the number of screw-keys required, to a few definite sizes. The arrangement of the mechanism, in the example now before us, is in the highest degree elegant and compact.

Fig. 1 is a side elevation, fig. 2 an end view, and fig. 3 a general plan of the entire machine.

The framing of the machine consists of a cast-iron head-stock A, A, in all respects similar to the fixed head-stock of the common turning lathe, as described in reference to the preceding Plate; in the present case, it is set upon a planed cast-iron table or bed B, B, supported at a convenient height by the pillars C, C. The cutting tool *x*, which is a hardened steel cylinder, serrated both on the face and edge, is fixed on the end of the main spindle *a*, running in bearings in the head-stock A, and adjustable by a back-centre in the same manner as in the ordinary lathe. A spur-wheel *w*, is fixed on the cutter spindle, and geers with the pinion *p*, on the driving shaft *k*, which also has its bearings in the head-stock, and carries the fast and loose driving pulleys *f*, and *g*. The motion for the self-acting feed of the machine, (which in this case is applied to the work and not to the tool), is derived from the shaft *k*, by means of a set of grooved pulleys marked *d*, of various diameters to suit the required velocity of feed, and connected by a cord to a corresponding set *e*, upon a longitudinal shaft *c*. On the opposite extremity of this shaft is an endless screw *s*, working into a worm wheel *h*, upon a transverse slide screw, which commands the motions of the sliding carriage on which the work is fixed; the worm wheel is capable of being slid upon its axis out of geer with the endless screw, by means of the handle *m*, in which case the slide-screw may be worked by the hand-wheel *b*.

The sliding carriage on which the nut to be operated upon is set, consists of two parts: a rectangular plate *i*, *i*, fitted to work smoothly and steadily in a rectilinear direction across the bed-frame of the machine, and guided by dove-tail pieces in the usual manner; and a circular table *l*, adapted to turn upon an axis in the centre of the plate *i*, and having six notches in its circumference, corresponding with the six faces of the nut to be dressed. This table is retained in the required position by means of a spring catch, the construction of which is distinctly shown in the plan fig. 3; the handle *n* serves to withdraw the catch when one face of the nut has been finished, and the table *l*, is then turned by hand until another of the notches is engaged by the catch and the adjacent face presented in its true position to the action of the machine, and so on with all the faces in succession.

The nut itself is fixed upon the travelling table by means of a mandril-pin or screw of corresponding dimensions, which is retained firmly in its place by a counter nut on the under side of a species of wrought-iron frame *r*, attached to the table *l*. Different sizes of these mandril-pins are of course required for the various sizes of nuts. The position of the cutter *x*, in relation to the work to be operated upon, is adjusted by means of the screw *y*, working in a nut fixed to the under side of the head-stock A, which is adapted to move longitudinally upon the table B, by having angular projections formed on its under surface, and fitting into corresponding grooves on the table.

PLATE XLIII.

PORTABLE DRILLING MACHINE, BY NASMYTH, GASKELL AND CO., MANCHESTER.

This machine differs essentially from that represented in Plates XXXIII. and XXXIV., in the mode of feeding the drill spindle. In Messrs. Whitworth's machine the feed is made self-acting, whereas in that now before us, it is obtained by the pressure of the foot of the man superintending the machine, which for small work is very convenient. Another peculiarity by which it is distinguished consists in its containing within itself the means of readily altering the speed of the drill spindle, according to the size of the hole to be bored; the two cone-pulleys by which this is effected being both situated in the machine itself.

Fig. 1 is a side elevation of the machine, and fig. 2 a front view. Fig. 3 is a partial section through the axis of the drill spindle. Fig. 4, a sectional plan taken immediately above the work table and shewing the form of cross section of the main framing. Figs. 5, 6, 7 and 8 are plans of some of the subordinate details, which shall be duly noticed in their places.

The framing A, A is formed of a single casting, its lower extremity being expanded into a broad flat sole, on which it rests on the floor of the workshop, without the aid of fixing bolts. The drill spindle B, B works through a short hollow shaft C, C, which at once serves to communicate the requisite rotatory motion to it, and to guide it vertically in ascending and descending. For this purpose the spindle is attached by a sunk feather to the interior of the tube C, which revolves in fixed bearings in the brackets D, D, bolted to the face of the framing, (see figs. 3 and 7); motion being imparted to it by means of the pair of bevil wheels *a*, and *b*, the former of which is fast upon the tube C, and the latter on the end of a horizontal shaft or spindle, having its bearings in the framing, and carrying the upper cone pulleys E. A corresponding set of pulleys, marked F, is similarly situated towards the lower end of the machine, and is connected with the upper set by a belt; being itself in the first instance set in motion by the fast and loose driving pulleys G and H, upon the outer end of its spindle.

The manner in which the feeding motion is communicated to the drill spindle is as follows: At the upper end of the machine a bracket *c*, is fitted to slide vertically on the face of the framing, being guided and retained in its place by dove-tail projections *g*, *g*, (fig. 5), working in a groove formed by the pieces *h*, *h*. To give motion to this sliding bracket, the end of a lever L, having its centre of motion in the framing, enters into a recess *d*, formed in the back of the bracket; a counter-balance weight W is suspended from its opposite extremity. This lever is connected by a rod I with another marked J, turning upon a centre formed in the socket K, bolted to the sole of the machine, (see figs. 1 and 8); this last is the treddle lever, by which the workman is enabled to convey the requisite pressure to the drill by applying his foot to its outer end. By this arrangement it is easy to perceive that when the foot is pressed upon the foot-board at J, the link I will cause the weighted end of the lever L to ascend, and the other to descend; and at the same time the sliding bracket *d*, into which is fitted the top of the drill spindle.

The mode of attaching the drill spindle B to the sliding bracket is rendered obvious by fig. 3; the top is formed with a collar upon it, which is retained in the interior of the boss *c*, of the bracket by the hollow screw *f*, bearing against the under surface of the collar. The spindle is at the same time acted on in the contrary direction by the screwed steel pin *e*, the end of which sustains the downward pressure when the foot is placed on the treddle J, which on being relieved resumes its original position by reason of the gravitating tendency of the weight W acting constantly to raise the drill spindle.

The table N, on which the work is set, is simply a wooden board or plank resting upon the top-plate of the bracket M. As the travel of the drill spindle is very limited, the table-bracket can be raised and lowered at pleasure, through the required range, by means of the screw S. Its sole is guided vertically by grooves *k*, *k*, in the main frame, which has also an opening *n*, and groove *l*, formed in it to receive the heads of the fixing bolts *m*, *m*, the nuts of which being screwed tight keep the bracket in its place. This arrangement will be clearly understood by reference to fig. 4.

PLATE XLIV.

PATENT RADIAL DRILLING MACHINE, BY JOSEPH WHITWORTH AND CO., MANCHESTER.

This is an entirely different arrangement of drilling machine from those before described. It embodies the elegant feeding apparatus introduced by the same makers, and described in the account already given of their vertical drilling machine, Plates XXXIII., XXXIV.; but in that form of the machine, the drilling spindle does not admit of any lateral motion; it is strictly confined to the same vertical position, and in that position can rise and fall at the will of the operator. But in the present case the drill spindle has not only the same vertical and revolving motions as in that form of the machine, but admits also of a lateral motion whereby it can be brought over the work, into any required position within the limits of the radial arm on which the whole drilling apparatus is carried.

For some special purposes the radial drill affords great convenience, but where much accuracy is required it cannot be so well depended upon as the ordinary form, as it rarely possesses the requisite degree of rigidity. It may be here observed, that the merit of first introducing this description of drilling machine is due to Messrs. Benjamin Hick & Son, of Bolton.

The arrangement consists of a strong upright framing A, A, with a sole by which it can be bolted to a stone foundation. To the face of this frame a vertical sliding bracket B, B, B is attached by dove-tail guides; this bracket may be raised and lowered at pleasure, according to the height required for the work, by means of a handle fitting on the end of the tangent screw L; this screw works into the worm wheel K, on the spindle of which is a small pinion geering with a rack on the back of the bracket B, B. At the upper and lower extremities of this bracket, bearings are provided for the gudgeons of the radial arm D, D; they are cast upon the ends of the sole C, C of the arm, and are situated at such a distance from the fixed framing as to admit of its traversing over an entire semi-circle. When set in the proper position over the work which is to come under the action of the machine, the arm is secured from shifting by means of a pinching screw and handle *w*, acting against the lower bearing.

It is to this arm D, D that the entire drilling apparatus is attached, the parts more immediately concerned in this operation being carried upon a moveable slide *b*, *b*, working upon planed dove-tail rails *a*, *a*, on the front of the arm; the slide *b*, is provided with an adjusting piece *c*, and set screws *d*, *d*, and may be set to any required position upon the arm by means of the slide screw *v*, *v*, passing through a nut fixed behind it in the usual manner.

Motion is imparted to this machine by means of the cone-pulley E. On the spindle of this cone is keyed the bevil wheel *f*, which geers with a similar wheel marked *e*, on the vertical spindle *g*, which is provided with a sunk feather to admit of its sliding through the eye of the wheel *e*, when the bracket is moved vertically. On the upper end of the spindle *g*, is keyed the bevil wheel *h*, which works into another similar wheel on the end of a horizontal hollow shaft G, having its bearing in the boss F. This hollow shaft has a groove cut inside of it to receive a feather inserted into the spindle *k*, which passes into it without being fixed, and thus admits of the longitudinal motion of the parts in relation to each other, while it prevents them from turning on their axes independently. The other end of the spindle *k*, has its extreme bearings in the slide *b*, and has the bevil wheel *o*, keyed upon it; this wheel geers with that marked *n*, on the drill spindle *l*. It is thus obvious that motion being communicated to the driving cone pulley E, it will be transferred, through the system of bevil geering above detailed, to the drill spindle *l*, in precisely the same way as we have described in reference to Plates XXXIII., XXXIV.

For the sake of simplicity, we have described the wheel which geers with that marked *h*, as being directly keyed on the hollow piece G; this, however, is not strictly true: the wheel is keyed on an independent short spindle of its own, which is also connected by a sunk feather with the intermediate piece G; so that the latter is nothing more than an adjustable sliding coupling between this spindle and that marked *k*, and is capable of being slid considerably further into the boss F than is represented in the drawing, fig. 1.

As already stated, the mode of feeding the drill spindle, and in fact all the drilling apparatus, is identical in arrangement with that represented in Plates XXXIII., XXXIV. Referring therefore to our description of that machine, we content ourselves, in this place, with a bare enumeration of the parts: *p*, *p*, are the friction pulleys on the same axes as the screw wheels which geer with the screwed part *r*, of the drill spindle. *q*, *q*, are the friction collars upon the pulleys *p*, *p*, and *s*, is the screwed rod by which the collars are brought into action. H is the hand wheel for raising or depressing the drill spindle by hand, I the back balance lever for neutralizing its weight, and *t*, the parallel motion for connecting its upper extremity with the lever I. *m*, is a boss formed on the lower end of the drill spindle, into which is inserted the socket for holding the drill itself.

PLATES XLV., XLVI.

BORING MILL AND LARGE TURNING LATHE.

In those establishments where steam engines of a large class are produced, machinery of corresponding dimensions and power is of course requisite for preparing and fitting the several parts for the functions which they have to perform. Of these parts the steam cylinder and its appendages are among the most important; much of the efficiency of the engine depending upon the accuracy with which they are constructed and fitted together. The interior surfaces of steam engine cylinders are dressed out by a machine called a *Boring Mill;* such machines are sometimes made so as to be applicable to no other purpose except that specified, but they are also frequently, and most judiciously, constructed with such modifications as to make them equally serviceable for fitting the pistons, and turning and polishing the cylinder covers, and every description of heavy articles.

The machine now under notice is of the kind last mentioned. It is simply a modification of the ordinary turning lathe, as represented in Plate XLI.; the principle of the *back-geer* arrangement being carried out so as to produce, with the least possible complication of mechanism, a very great range of velocity, so as to adapt the machine for the various purposes for which it may be employed.

Fig. 1 is an elevation, and fig. 2 a plan of the machine, as adapted to bore out a cylinder of considerable dimensions. Fig. 3 represents the frame to which the boring tools are attached, and fig. 4, an essential part of the self-acting geering by which it is made to traverse longitudinally through the cylinder.

General Description.

The lathe-head L, in which the bearings of the driving spindles are situated is in no way distinguished from the ordinary form, except for its unusual strength and breadth; being adapted to sustain three shafts instead of two. It is firmly bolted to a foundation of masonry, of such height as to raise it to a sufficient distance from the floor to admit of the largest object which may at any time be subjected to the action of the machine. A massive cast-iron sole-plate B, B, also embedded and secured to a stone foundation, serves to give the requisite firmness and rigidity to the parts sustaining the strain of the machine; numerous longitudinal slots are formed in this bed-plate, to allow of the introduction of bolts, in any required position, for fixing the work, or the tools which may be employed to operate upon it. When the machine is used as a boring mill, (as shewn in the Plate), the cylinder S, S, which is to be subjected to its action, is clamped down to the bed-plate by strong bars *t*, *t*, and the boring bar A, A is carried in brackets C, C and D, D, also securely bolted to the same sole, and guided into their proper positions by the planed rail *s*, *s;* but in case of it being employed as an ordinary turning lathe, the plate B serves as a foundation for the brackets required for supporting the shifting head-stock and the slide rest.

The driving cone-pulley F, with its attached pinion *a*, is fitted to run loose upon the main spindle H, which is supported by a back centre, and provided with a great chuck or face-plate G, as in ordinary turning lathes. Upon the back of the face-plate, which is perforated with numerous bolt holes for the purpose of fixing the work to be turned, is the great internal spur wheel *k*, *k*, by means of which all the slower motions are communicated to the lathe. The arrangement of intermediate geering between this spur-wheel and the pinion *a*, on the cone-pulley is as follows:—Two horizontal shafts *o*, and *p*, (which may be called respectively the second and third motion shafts), are fitted to revolve in suitable bearings in the lathe-head L; the shaft *o* carries a fixed spur-wheel *b*, which geers constantly with the pinion *a*, on the cone-pulley; on the adjacent end of the shaft *p*, is a wheel *c*, sliding upon a sunk feather so as to be capable of being thrown into geer with the wheel *b*, or disengaged at pleasure. On the same shaft is a wheel *d*, working into a pinion *e*, on the second motion shaft; this pair also admits of being slid out of geer when required. Finally, the shaft *p*, carries on its inner extremity a small pinion *i*, working into the teeth of the great internal spur-wheel *k*, *k*, on the back of the face-plate.

For producing the highest degrees of speed of which the machine is susceptible, the arrangement of geering is precisely the same as in the ordinary back speed lathe; the pinion *g*, upon the second motion shaft *o*, admitting of being engaged with the wheel *h*, on the lathe-spindle, by sliding the latter upon a sunk feather towards the face-plate. In this case it is, of course, necessary to remove the pinion *i* from contact with the internal wheel *k;* for which purpose the bearings of the shaft *p*, are made capable of sliding inwardly to the required distance, when they may be secured by shifting the small keys *f*, *f*, *behind* them.

Mechanism of the Boring Bar.—The boring bar A, A is a massive and rigid cast-iron shaft, turned to a truly cylindrical form throughout its entire length; its extremities are supported in bearings in the standards C, C and D, D, and motion is communicated to it from the lathe by a coupling, *y*, bolted to the face-plate. The boring-block E, which carries the cutter frames, is very accurately fitted to slide longitudinally upon the bar, in obedience to the motion of the feeding screw, which is sunk into a recess formed in the bar, (see fig. 1), and works through a nut inserted into the boring-block at *v*. The mechanism employed for producing the self-acting motion of the feed screw is extremely simple and ingenious: Upon the outer extremity of the boring-bar is fixed a spur wheel *m*, and on the opposite side of the bearing a similar wheel *l*, of exactly the same number of teeth, is fitted to turn loosely upon it; this last wheel is formed with the same number of internal as of external teeth, the internal teeth geering with a small pinion fixed upon the end of the feed screw, (see fig. 4). These wheels are connected together by a short shaft *u*, carried in bearings in a small adjustible bracket *r*, bolted to the side of the standard D; the pinions *n*, and *q*, working into the wheels *m*, and *l*, respectively. The number of teeth in the pinion *q*, is less than that in the pinion *n*, by one; so that one revolution of the shaft *u*, causes the wheel *l*, to advance one tooth on the boring bar, and consequently produces a corresponding amount of angular motion upon the feed screw. The pitch of the screw being ½ inch, and the small pinion fixed upon it having 16 teeth, the feeding motion of the boring-block will be ·5 ÷ 16 = ·03125 inch for each turn of the shaft *u*, or $\frac{\cdot03125 \times 64}{35} = \cdot0571$ inch during one revolution of the boring bar; 64 being the number of teeth in the wheels *l* and *m*, and 35 that of the pinion *q*.

The circular cast-iron frames *w*, and *x*, for carrying the boring tools, are shewn in elevation in fig. 3, and in section in their proper position in the machine, at fig. 1. It is found most convenient to employ two of these; the internal ring *w*, which is of a diameter suitable for boring the smallest sized cylinder for which this machine would be used, is fixed permanently upon the boring-block E, and serves as a bearing surface for attaching by means of keys other frames, of diameters corresponding to those of the cylinders to be operated upon. The frame *x*, is of a size suited to bore the cylinder S, S; it is formed with 24 slots in its periphery; 12 of these receive the cutters *j*, *j*, which are adjusted and fixed in their places by thin wedges; the remaining slots are fitted with pieces of hardwood driven firmly into them, for the purpose of steadying the motion, and presenting a general guiding surface.

Details of Geering and Calculations of Speeds.—In order at once to shew the great range of velocity of which a lathe constructed upon this plan is susceptible, and to illustrate the mode of determining the dimensions of the various pulleys, wheels, and pinions in machines of this nature, the following tables and calculations are appended:—

TABLE OF SPEEDS WHICH MAY BE OBTAINED IN THIS LATHE.

No.	Revolutions per minute.		No.	Revs. per minute.
1	·333 × 3 = 1		13	4·839
2	·416		14	6·049
3	·520		15	7·561
4	·650	$\sqrt{\cdot650} \times 3 = 1\cdot4$	16	9·451
5	·812	$\sqrt{\cdot812} \times 3 = 1\cdot56$	17	11·814
6	1·015		18	14·767
7	1·269		19	18·457
8	1·586		20	23·079
9	1·982	1·982 × 3 = 5·946	21	28·842
10	2·478		22	36·053
11	3·077		23	45·066
12	3·871		24	55·333

These numbers form a progressive series, the ratio of increase being as 1 to 1¼, so that any speed within the range may be procured to within ⅛ of that required; thus supposing the required surface speed of the cutters to be 7 feet per minute, the greatest deviation will be $\frac{84}{8} = 10\frac{1}{2}$ inches per minute.

The cone pulleys of the machine are driven by a similar set of cone pulleys on an intermediate shaft. This shaft is again driven from the main shaft by pulleys of the following relative diameters and disposition:

3 feet.	22½ in.
22½ in.	3 feet.

The diameters of these pulleys are to each other as the first to the fifth speed of the bar, so that the smaller is to the larger pulley as $\sqrt{\cdot333} : \sqrt{\cdot812}$, or as 1 : 1·56. The increase of speed from the largest to the smallest pulley on the spindle H is as the first to the fourth speed, and the diameters of the pulleys are in the ratio of $\sqrt{\cdot333} : \sqrt{\cdot650}$, or 19 : 26½.

The first eight speeds are obtained with the wheels *d*, and *e*, the second eight with the wheels *b*, and *c*, and the third eight by geering *g*, and *h*, disengaging *e*, and *c*, and taking the pinion *i*, out of geer with the large internal wheel on the face-plate by shifting the shaft *p*, towards the shaft *o*.

NUMBERS OF TEETH IN WHEELS.

Wheels of the Driving Geer.	No. of teeth.	Wheels of the Feed Geer.	No. of teeth.
a,	24	*l*, external,	64
b,	52	*l*, internal,	64
c,	40	*m*,	64
e,	14	*n*,	36
d,	64	*q*,	35
g,	34	Pinion on traverse screw,	16
h,	40		
i,	15		
k,	144		

The speeds produced by the wheels *e*, *d*, and *b*, *c*, are to each other as the first to the ninth speed of the chuck; therefore, $\frac{64 \times 52}{14 \times 40} = 5\cdot94$ nearly.

The proper velocity of the cutting tools in boring being about 7 feet per minute, the slowest speed, namely, ⅓ of a revolution per minute, is suitable for boring a cylinder of 80 inches in diameter; for $\frac{84 \times 3}{3\cdot1416} = 80$. All the cylinder boring speeds are in the first eight of the table—the others being employed for turning and polishing heavy articles, such as large cylinder covers, &c.

PLATES XLVII., XLVIII.

BORING AND TURNING MACHINE, BY KINMONDS, HUTTON AND STEEL, DUNDEE.

These drawings exhibit very fully the general mechanism of an exceedingly elegant and useful machine, adapted to the several purposes of boring, cylindrical turning, and facing. It fulfils completely the double purpose of the large lathe and boring-mill, and moreover, possesses some important peculiarities which entitle it to a high rank in the order of machines to which it belongs.

Fig. 1 is a side elevation of the machine.

Fig. 2 is a general plan corresponding.

Fig. 3 is an end view from the back of the fixed head-stock; and

Fig. 4 is a transverse section taken in front of the shifting head-stock.

General Description.

The working parts of the machine are placed upon a substantial bed-plate A, A, having projections cast upon its edges at intervals, for the purpose of bolting it to a stone foundation. The form of the cross section of this bed-plate is shown in fig. 4, as also the mode in which the shifting head-stock is retained upon it. The fixed head-stock B, B is provided with four bearings, cast in each of the two standards, for the purpose of receiving the cone spindle C, the second motion shafts D and E, and the main spindle F, upon which the face-plate G is fixed. The geering of the machine is calculated to produce a series of variations in the speed of the main spindle F, independently of any variation which may be effected by means of the driving cone-pulley *a*.

To effect this, the arrangement of the wheels upon the shafts D and E is such that either of them may be thrown into geer with the cone spindle C, and the internal wheel *g*, on the back of the face-plate G. Thus, if the wheel *e*, on the shaft D, be brought into geer with the wheel *d*, on the cone spindle, the pinion *f*, will at the same time be in geer with the internal wheel *g*, and a quick motion will be communicated to the face-plate; but if the opposite shaft E be slid forward longitudinally till the wheel *j*, geers with the pinion *h*, the pinion *f′*, on that shaft will be thrown into geer with the internal wheel, and a slower motion will consequently be imparted to the face-plate. Again let both the shafts D and E be thrown out of action; the cone spindle C may then be directly connected with the main spindle F by means of the wheels *b*, and *c*, the relative sizes of which may be varied so as to produce any required velocity; this latter arrangement is only employed for obtaining a high speed, as in the case of polishing.

The loose head-stock H is adjustable to any required position by means of a crank-handle fitting upon the square end of the spindle marked *o*, which communicates by a train of toothed geer with the rack M, fixed upon the bed-plate of the machine, as shown in fig. 4. The pinion which works into the rack is keyed upon the spindle *p*, carried in bearings attached to the sole of the head-stock. On the same spindle is a small bevil wheel, which geers with a pinion on the lower end of a vertical spindle, having its bearings in the interior of a hollow column, cast in the body of the head-stock. On the upper end of this spindle is another bevil wheel, which geers with a pinion on the horizontal spindle *o*, thereby completing the connection with the fixed rack M. This arrangement is fully exhibited in dotted lines in fig. 4.

The same rack M serves also for moving the slide-rest K in a longitudinal direction, by means of a pinion keyed upon the shaft *q*, shown in fig. 2. This shaft is carried in bearings attached to the edge of the sole-plate of the slide J, and terminates in a *square*, to which a lever may be applied to give motion to the shaft. The sole-plate of the slide is provided with dove-tail grooves in its under surface, to receive the correspondingly formed heads of two bolts, for the purpose of attaching and fixing to the saddle-plate J, the sole of the bracket K, which carries the slide-rest L. To afford the utmost possible facility for adjustment, the bolt holes in the sole of the bracket K are slots of considerable length, and the fixing bolts hold the two plates firmly together, metal to metal, their surfaces of contact being planed true.

By this arrangement the slide may be made fast in any position upon the saddle-plate, which in turn is retained upon the bed-plate of the machine by wedge-pieces worked by means of two horizontal eccentric spindles *v*, *v*, shown in fig. 2.

The longitudinal and transverse motions of the tool-holder, for the purposes of traversing the work, and placing the tool in and out of cut, are obtained by means of the screws *r*, and *s*, which work at right angles to each other, in the usual manner; the tool is made fast on the tool-holder by means of the two clamps *t*, *t*. In adjusting the slide primarily to the work, the sole K is released from the saddle-plate J by relaxing the connecting bolts; the bracket K is then shifted to the required distance from the longitudinal axis of the machine, and also, to a certain extent, in the line of that axis by sliding the bolts in the dove-tail grooves of the saddle-plate, should that operation be more convenient than moving the latter on the bed-frame of the machine. For transverse adjustment to a limited extent, the screw *s*, can be used; for longitudinal adjustment, the tool-carrier may be set in a similar manner by the screw *r*.

Action of the Machine.—Supposing it is required to *face* a heavy piece of work by this machine, it is clamped to the face-plate by means of bolts which pass through radial slots formed in the latter. The wheel *c*, upon the end of the main spindle F, is then removed, and the shaft E is slid longitudinally in its bearings until the wheel *j*, and the pinion *f′*, both keyed upon it, geer respectively with the pinion *h*, upon the cone spindle, and the internal wheel *g*, which is fast upon the back of the face-plate. By this arrangement the slowest motion of the face-plate is obtained. If a quicker motion be required, as when the action of the tool is near the axis of the machine, the shaft E is thrown out of geer, (as shown in the views, figs. 1 and 2), and the shaft D is moved endways until the wheel *e*, and pinion *f* upon it, geer respectively with the wheel *d*, upon the cone spindle, and the internal wheel *g*, upon the back of the face-plate. The speed is thus increased in the ratio of the number of teeth in the wheel *d*, and pinion *h*; that is, as 51 to 15. A still higher speed, and indeed the highest, is obtained by arranging the geering of the machine as it is represented in the engravings. The shafts D and E, it will be observed, are both out of geer, (being retained in that position by the catches *k*, *k′*), and the wheel *b*, upon the cone spindle C, is in geer with the wheel *c*, upon the end of the main spindle F, so that the speed of the cone is transmitted to the face-plate through the single pair of wheels *b*, and *c*, which are to each other in the ratio of 51 to 66.

These three speeds, which are independent of the five speeds obtained by the cone, may be thus compared:—The numbers of teeth in the wheel *j*, and pinion *f′*, upon the back shaft E, are respectively 78 and 13, and the numbers in the pinion *h*, upon the cone spindle, and in the internal wheel *g*, upon the back of the face-plates, are 15 and 119; consequently when the shaft E is in geer, the ratio of the speed between the cone spindle and the face-plate is as 78 × 119 : 13 × 15, or as 47·6 to 1, being the slowest motion of which the machine is capable. Again, the numbers of teeth in the wheel *e*, and the pinion *f*, upon the shaft D, are respectively 51 and 13; and the numbers in the wheels *d*, upon the cone spindle, and the internal wheel *g*, being 51 and 119, therefore when the shaft D is in geer, the ratio of the speed of the cone spindle to that of the face-plate is as 119 : 13 or as 9·15 to 1. And when both of these shafts are out of geer, and the wheel *b*, upon the cone spindle is working into the wheel *c*, upon the main spindle, the numbers of teeth being respectively 51 and 66, the ratio of the speed is 1 to 1·3 nearly.

The action of the machine in ordinary parallel turning is the same as in any common lathe. The mode of obtaining a self-acting longitudinal motion of the tool-carrier is by a stellar plate fixed upon the end of the screw *r*, and which is worked by an arm bolted to the face-plate or to the object which is being turned, so as to come in contact with the plate, and cause it to advance one tooth at each revolution.

Application of this Lathe to the Boring of Cylinders.—When the machine is to be used as a boring-mill, the slide-rest and shifting head-stock are removed, and a boring-bar, of which the annexed cuts show the details, is substituted; one end being supported by a standard fixed upon the bed-plate, in the same manner as shown in Plates XLV.—XLVI., while the other is made fast to the face-plate. The end *c*, of the bar enters the hole formed for the reception of the running centre of the lathe spindle, and the two projecting pieces *d*, *d* pass into two of the radial slots of the plate, so that the bar will of necessity have the same speed as the face-plate. The bar *a*, is recessed on one side to receive the screw *m*, by which the cutter-block *b*, is traversed longitudinally upon it. This block is provided with a nut which enters the recess of the bar, and through which passes the screw *m*, as shown in fig. 3 of the annexed cuts. The mechanism for obtaining the feed-motion, although identical in principle with that we have already explained in the preceding description, differs considerably in the details of its construction; it is represented generally by fig. 1, and more fully by the sectional view fig. 2. A small spur wheel *l*, is keyed upon the end of the screw, and geers with a wheel *k*, of the same diameter, on the spindle *j*, the end of the boring-bar being formed with a cylindrical recess of the same diameter as the wheel *k*, which passes within it, and is retained in its place by a solid bush of metal bored to fit the spindle *j*, and to serve as its bearing. A spur wheel *e*, of small pitch, is fixed upon the projecting end of the bush in which the spindle *j*, works, and which is attached immovably to the boring-bar by means of a screwed key passing through the metal of the wheel and between the surfaces of contact of the boring-bar and the included bush. Another wheel *h*, differing slightly in the number of its teeth from the wheel *e*, is keyed upon the end of the spindle *j*, but is otherwise independent of the bar. These two wheels are made to geer with two pinions, marked *f*, and *g*, in fig. 1, both keyed upon the same fixed axis, or to each other. Now the wheel *e*, being fast to the bar and revolving with it, will communicate its motion to the pinion *f*; but the pinions *f*, and *g*, being attached together, the motion of the wheel *e*, will be communicated by *g*, to *h*, and through the spindle *j*, to the wheel *k*, which is in geer with the wheel *l*, upon the end of the feeding screw *m*, and which will thus receive a motion proportional to the difference of the wheels *e*, and *h*. Thus, supposing the wheel *e*, to have 71 teeth, and *h*, to have 70, then the bar having made 71 revolutions, the screw will have made one revolution, and advanced the cutter-block through a space equal to the pitch of the screw. By making the wheel *h*, larger than the wheel *e*, a contrary motion would be given to the screw, since in that case the wheel *h*, would be driven more slowly than the wheel *e*; and if both wheels had the same number of teeth the screw would obviously have no motion on its own axis. This is what is termed in mechanics a *differential motion.*

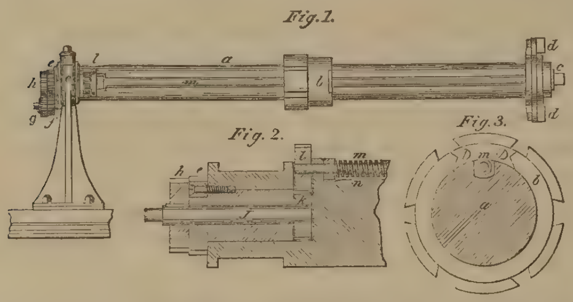
Fig. 1.

Fig. 2.

Fig. 3.

Literal References.

A, A, the bed-plate of the machine.
B, B, the fixed head stock, bolted to the bed-plate.
C, the driving cone-spindle.
D, E, the second motion shafts.
F, the main spindle carrying the face-plate G.
a, the driving cone-pulley with five speeds.
b, a wheel of 51 teeth working into
c, a wheel of 66 teeth on the main spindle.
d, a wheel of 51 teeth working into
e, an equal sized wheel on the second motion shaft D.
f, *f′*, pinions of 13 teeth on the shafts D and E, working into
g, the internal wheel of 119 teeth attached to the face-plate.
h, a pinion of 15 teeth working into
j, a wheel of 78 teeth upon the second motion shaft E.
k, *k′*, catches for retaining the shafts D and E when put in or out of geer.
l, *l*, stay-rods for strengthening the fixed head-stock.
H, the shifting head-stock.
m, a screw-spindle with hand-wheel for adjusting the centre in the shifting head-stock.
n, a pinching screw for fixing the centre when adjusted.
o, a spindle for moving the shifting head-stock longitudinally.
p, a transverse shaft forming part of the mechanism by which the shifting head-stock is moved.
h′, *h′*, hooked bolts for fixing the shifting head-stock.
J, the saddle-plate, forming a support for
K, a bracket for carrying the slide-rest.
L, the longitudinal carriage of the slide-rest.
M, the toothed rack, fixed to the bed-plate for the purpose of moving the slide-rest and shifting head-stock.
q, a shaft carrying a pinion which works into the rack M, for moving the slide-rest longitudinally.
r, longitudinal screw of the slide-rest.
s, transverse screw of do.
t, *t*, clamps for fixing the tool upon the slide-rest.
u, screw for fixing the slide-rest.
v, *v*, screws for fixing the saddle-plate.

PLATES XLIX., L.

SMALL SELF-ACTING AND SCREW CUTTING LATHE,

BY CHARLES WALTON, LEEDS.

These Plates represent, in full detail, the mechanism of a very efficient and in every respect convenient turning lathe, adapted for the smaller descriptions of work, in which great accuracy is required. It is peculiarly suitable for cutting the threads of long screws, such as are frequently employed in the construction of tools of the same kind as itself; it is self-acting in all its movements, and capable of having these modified in an infinite variety of ways, in order to suit the nature and circumstances of the work under its operation.

Enumeration of the Figures.

Fig. 1 is a general side elevation of the lathe, and fig. 2 is a plan corresponding.

Fig. 3 is an end elevation showing the geering.

Fig. 4 is a transverse section taken between the fixed head-stock and the slide-rest, showing the latter in elevation, as also the arrangement of the geering for causing it to travel longitudinally and transversely.

Figs. 5, 6, 7, and 8 show details of the geering for working the slide-rest.

Fig. 9 is an elevation of the upper cone and driving pulleys; these consist of two sets, the smaller being used for reversing the motion of the saddle when the lathe is employed in screw cutting, and the larger when the tool is in action, and a slower motion consequently is necessary.

Fig. 10 is a section through the driving cone on the lathe-spindle.

Fig. 11 is a front view of a spiral expanding chuck, adapted for this lathe.

Fig. 12 is a side elevation of the same; and

Fig. 13 a vertical section in the plane of the lathe-spindle.

General Description.

The machine is carried upon three standards A, A, A, of which the general forms are shown in figs. 3 and 4. These standards are planed on their upper surfaces to afford a solid support for the bed-frame B, B, the upper surface of which is also planed. The exterior edges of the bed are bevilled in the usual way, for the purpose of retaining the saddle-plate L, L of the slide-rest, as shown in the cross section, fig. 4. The fixed head-stock C, C, is bolted firmly to the bed-plate and carries the main spindle D, upon which is the driving cone a, a section of which, showing its relation to the spur wheel e, and pinion b, forms the subject of fig. 10. The cone is, as usual, loose upon the spindle, but may be attached at pleasure to the wheel e, which is fast upon the spindle, when it is necessary to throw the back-speed shaft E out of geer. This is effected by the hand-rail G, which connects the two levers commanding the bearings of the shaft in the two projecting pieces F, F of the fixed head-stock; a method commonly adopted when the arrangement of the geering does not conveniently admit of the shaft being shifted longitudinally.

In screw cutting, the motion of the leading or guide screw N is derived from the cone spindle through the train of wheels v, w, x, y, z, the relative dimensions of which admit of being varied indefinitely. In plain work the longitudinal motion of the tool is obtained through the the train v, a', d', e', and the band pulleys b', and c, for varying the speed; these convey the motion of the main spindle to the traverse spindle f', f', which, by means of the endless screw g', (figs. 4 and 8), and worm wheel i', communicates through the intervening spur pinions r, and s, on the cross shaft q, with the pinion t, (fig. 5); this pinion geers with the toothed rack u, (figs. 4 and 7), attached to the under side of the saddle-plate L of the slide-rest. This circuitous mode of conveying the motion is resorted to in order that the longitudinal rack u, may be fixed to the bed frame in an inverted position, so as effectually to preserve the teeth from being clogged by dirt or pieces of metal which would otherwise impede or prevent its action.

The geering for reversing the motion of the saddle consists of three meter wheels and the clutch-box k', arranged upon the traverse shaft f', f'. The clutch k', communicates with the reversing lever l', in front, by means of a forked rod fixed upon a horizontal shaft, passing through the bed of the lathe. Thus the shaft communicating with the train of wheels from the main spindle may be connected either directly with the traverse rod f', f', or through the intervention of the meter wheels when it may be necessary to change the direction of its motion. A weighted lever j', shown in fig. 4, serves the purpose of throwing the worm-wheel i', in or out of geer with the endless screw g', upon the traverse rod, thereby connecting or disconnecting the lathe with the saddle of the slide-rest as may be required. The slide-rest can be relieved from connection with the leading screw N by means of the handle o, attached in front of the saddle; by pressing this handle down, it acts upon a stud in the plate carrying the screw-box n, which is thereby opened, and the saddle relieved.

The sole of the movable head-stock J, J does not rest immediately upon the bed of the machine, as in ordinary lathes, but is fitted upon an intermediate piece K, upon which it is capable of sliding transversely to a certain extent, a screw f, being provided for the purpose of regulating its position. This simple arrangement adapts the lathe for conical turning.

Action of the Lathe.—The arrangement of the geering in the views given of the lathe in the Plates, is that adapted to screw cutting. The cone a, which is loose on the spindle, is fast to the pinion b, of 13 teeth; this pinion geers with the wheel c, of 52 teeth, upon the back-speed spindle E, which also carries the pinion d, of 13 teeth, geering with the wheel e, of 52 teeth, fast upon the cone spindle D. According to this arrangement, the ratio of the speed of the driving cone to that of the main spindle is as 16 to 1.

The connection between the cone spindle and the leading screw N is accomplished by means of the pinion v, of 40 teeth, fast upon the driving cone spindle; this wheel is working into the wheel w, of 60 teeth, upon a stud passing through a circular slot in the bracket O, bolted upon the fixed head; this latter wheel again is in geer with the wheel x, of 90 teeth, also upon an adjustable stud, and carrying a pinion y, of 45 teeth, in geer with the wheel z, of 90 teeth, fast upon the leading screw shaft N. This train can, of course, be varied at pleasure to suit the particular pitch of screw to be cut, the slots in the bracket O being so formed as to admit of the studs of the carrier wheels being shifted to allow them to come into geer.

Literal References.

A, A, A, the standards upon which the lathe is supported.
B, B, the bed or shears of the lathe.
C, C, the fixed head-stock bolted to the bed.
D, the main spindle revolving in conical steel collars in the fixed head-stock.
a, the driving cone of the lathe.
b, the first pinion, of 13 teeth.
E, the back-speed shaft revolving in bearings in the projecting pieces F, F, of the fixed head-stock.
c, wheel on the back speed shaft E.
d, pinion on do.
e, wheel fixed on the main spindle.
G, hand-rail and levers for throwing the back-speed shaft into and out of geer.
H, the face-plate screwed to the main spindle.
I, the back centre bracket.
J, J, the movable head-stock fitted upon
K, a saddle resting upon the bed-plate, and formed so as to admit of the head-stock moving transversely upon it.
f, a screw for setting the loose head-stock transversely.
g, the hand-wheel of the loose head-stock; and
h, a handle formed into a nut, for tightening its pinching screw.
L, L, the saddle-plate of the slide rest.
M, the tool-holder of the slide rest.
i, adjustable piece by which the slide-rest is retained upon the saddle-plate.
j, the rest-plate of the tool-box.
k, screw for fixing the tool-holder upon the slide-rest.
l, hand-wheel for working the transverse screw of the slide-rest.
m, crank-handle for setting the tool into and out of cut.
N, the leading or guide screw of the lathe.
n, the screw-box for the leading screw.
o, a handle for connecting and disconnecting the screw-box.
p, the crank-handle for working the saddle-plate by hand.
q, the transverse shaft of the slide-rest.
r, a pinion upon the shaft *q*, working into
s, a small spur-wheel working under the saddle-plate.
t, a spur pinion on the same spindle as *s*, and which geers with
u, a rack fixed to the bed of the lathe.
v, the first pinion in the trains of the head-geering of the lathe.
O, bracket for carrying intermediate wheels, by which the motion of the main spindle is transmitted to the leading screw.
w, *x*, *y*, *z*, trains of intermediate or carrier-wheels.
a', b', c', d', e', geering for conveying the motion of the main spindle to
f', f', the traverse rod or worm-shaft.
g', a worm or endless screw upon the traverse spindle, adapted to slide longitudinally upon it, and geering with
i', the worm-wheel on the end of the transverse shaft *q*.
j', a weighted lever for disconnecting the worm-wheel i'.
k', the reversing geer upon the traverse shaft f', f'.
l', the lever for regulating the reversing geer.
P, figs. 11, 12 and 13, the front plate of the universal chuck.
Q, the back plate of the same, showing the spiral groove for expanding and contracting the clutches or jaws.
R, R, R, the clutches or jaws of the chuck. They are each formed with two strong pins projecting inwardly, so as to fit into two contiguous threads of a spiral groove cut in the back plate Q; and are constrained to move in a rectilinear direction by radial slots formed in the front plate P. When the back plate is turned upon its axis, which coincides with the axis of the main spindle, the front plate being meantime held fast, the clutches or jaws will be guided simultaneously, further from, or nearer to, the centre, and thereby be made to grasp the work in the usual way. The advantage of this arrangement over the mode of working the clutches by separate screws, consists in their being simultaneously expanded and contracted in respect of the centre.

PLATE LI.

PUNCHING AND SHEARING MACHINE, BY CAIRD AND CO., GREENOCK.

These drawings represent the form and general arrangement of a machine of great importance and utility in the manufacture of steam engine boilers. The present example is distinguished for its mechanical elegance of design, simplicity of construction, compactness and strength. Although the machine occupies only a very inconsiderable space on the floor of the factory, it is capable of punching and shearing plates of one inch in thickness.

Fig. 1 is a general side elevation of the machine.

Fig. 2 is a front elevation, looking upon that face of the machine which is adapted to the operation of shearing.

Fig. 3 is a corresponding elevation of the opposite end at which the operation of punching is performed.

Fig. 4 shows the form of the main shaft, and section of the slides.

General Description.

The framing consists of a single massive casting A, A, having strong brackets B, C, and D, E formed upon it at opposite sides. In the pieces B and D recesses are formed for the reception of the bushes of the shaft N, and of the slides *b*, *b*, to which the shearing cutter and punch are attached respectively. The bushes of the shaft N are adjusted to the proper degree of tightness by the cotters *d*, *d;* and the extremities *a*, *a*, of the shaft, close to these bearings, are formed eccentrically, as shown in fig. 4. These eccentric ends are inserted into the slides *b*, *b*, and work in oblong bushes of such a form as to allow the eccentrics to move freely in a lateral direction, while the full amount of their vertical motion is transferred to the slides. These bushes are retained in their places by thin wrought-iron covers *c*, *c*.

The slides move vertically between the parallel dove-tail guides *e*, *e*, fixed by screws tapped into the projecting pieces B and D, which are carefully dressed to allow the slides to move freely but without play upon them. On the shearing slide is fixed a steel cutter *f*, acting in contact with the stationary cutter inserted into the table C; the cutting edges form an acute angle with each other, so that during the process of shearing, the action is rendered gradual. The opposite slide carries the punching tool *g*, which is held in its socket by a cotter *h;* the small hole *i*, immediately over it, is for the convenience of driving the punch out of the socket when required. The die holder is attached to the table E by two screws tapped into the table, and thus admits of being changed at pleasure.

The shaft N derives its motion from the large bevil mortice wheel M, M, keyed upon it between the cheeks of the frame. This wheel geers with, and is put in motion by, the pinion L, on the lower end of the vertical shaft J, which is carried in a step supported in a bracket cast on the inside of one of the cheeks; the upper end revolves in an independent bearing attached to any convenient beam. The power is transmitted to this shaft from the driving-shaft F, by means of the two bevil wheels G and H. On the upper end of the same shaft J is keyed the fly-wheel K, for equalizing the motion of the machine under the irregular strains to which it is subject.

Action of the Machine.—Motion being communicated to the eccentric shaft N, the slides will be made to travel vertically through spaces corresponding to the eccentricity of the parts *a*, *a*, thereby working the shears and punch alternately; the eccentricity of the two extremities being formed on opposite sides of the shaft, so that while the punch is descending, the cutter of the shears will be ascending, and *vice versa*. The plates under operation are shifted by hand, upon tables of wood erected at the proper levels, and usually with guides fixed upon them for ensuring accuracy in the operation of cutting.

Literal References.

A, A, the frame of the machine.
B, hollow bracket for the shearing slide.
C, the fixed table for the same.
D, hollow bracket for the punching slide.
E, the table upon which the hollow die is set.
a, *a*, eccentric ends of the shaft N.
b, *b*, the shearing and punching slides.
c, *c*, covers fixed upon the slides, over the ends of the eccentrics *a*, *a*.
d, *d*, cotters for adjusting the adjusting bearings of the shaft N.
e, *e*, dove-tail guiding pieces between which the slides move.
f, the shearing cutter.
g, the punching tool.
h, a cotter for fixing the punch in its socket.
i, an oblong hole over the socket of the punch, for driving it out when required.
F, the shaft by which the power is led to the machine.
G, a bevil wheel on the horizontal driving shaft, geering with
H, a bevil wheel on the vertical driving shaft J.
K, the fly-wheel for regulating the motion of the machine.
L, a bevil pinion on the vertical shaft J, geering with
M, a large bevil mortice wheel fixed on
N, the main eccentric shaft.

PLATE LII.

IMPROVED RIVETING MACHINE, BY WILLIAM FAIRBAIRN AND CO., MANCHESTER.

In the manufacture of steam engine boilers, however varied and important the improvements which have, from time to time, been effected in the form and arrangement of their parts, no attempt has, until a very recent period, been made to facilitate the means of their construction, or, by the introduction of machinery, to supersede the necessity for manual labour. It is true, the punching and shearing machine has, under various modifications, been long in use, but it is only within the last few years that machines for bending plates, for making rivets, and still more recently for *riveting*, have been introduced.

For this last purpose, a variety of ingenious and effective combinations have been proposed, and although, as yet, none of them has come into very general use among boiler-makers, there can be little doubt that the laborious and expensive process of riveting by hand will eventually be superseded by some form of this machine. The first idea of the riveting machine is due to Mr. Fairbairn, of Manchester, who, in 1838, patented a machine in many respects similar to the common punching machine, but having the great lever of such a form as to communicate a horizontal motion to the dies or tool for forming the head of the rivet. The machine represented in Plate LII. is a modification which Mr. Fairbairn has since made, in which he has introduced several improvements, and remedied several defects to which the former was subject.

The principle of Mr. Fairbairn's machine consists in its performing, by almost instantaneous pressure, what could formerly only be done by a long series of impacts. Every mechanic is aware that the operation of riveting, in all ordinary cases, requires the services of three men, one to hold a hammer or other mass of iron inside the boiler, against the head of the rivet, while the other two beat the protruding end into the conical form given to the rivet on the outside of the boiler. For this operation very expert and skilful workmen are required, in order that the rivets may be fixed soundly and firmly without injury to the plates, and that all unnecessary hammering, which has only the effect of weakening the rivets, may be avoided. By means of the riveting machine, the process is accomplished with much greater rapidity and regularity, without producing the stunning and disagreeable noise unavoidable in hand-riveting. Besides these advantages, the operation being, as we have before said, performed almost instantaneously by the machine, the danger of injuring the rivets by hammering them when too cold is avoided, and the hemispherical, which is doubtless greatly preferable to the conical form, is more easily impressed upon them.

Fig. 1 represents an elevation, and fig. 2 a plan of Mr. Fairbairn's machine in its most improved form, and as it is now constructed by him. It possesses the advantage over his first proposed form, of being more compact and portable, and is capable of more extensive application, being adapted to rivet angle-iron, and finish the *corners* of boilers and cisterns.

General Description.

The sole or base A, A of the machine is made of cast-iron, and mounted upon wheels adapted to rails, for the convenience of shifting it to any required place. The framing B, B is cast in a piece with the sole A, and consists of an oblong box, open at the top, and furnished with bearings for the movable parts of the machine. A strong upright stem C, of malleable iron, is fitted firmly into the base A, which is secured against the effect of undue strains, (arising from the dies coming into contact with a cold rivet, or other hard substance), by a wrought iron strap D, passing round its upper edge, and secured by nuts at *a*, *a*. The stem, or riveting block C, is the point of resistance to the action of the dies, and against it is placed that part of the boiler which is to undergo the process of riveting. It is made of malleable iron, in order that it may possess a certain amount of elasticity, which is necessary to the prevention of such accidents as we have just alluded to. Its upper extremity is formed into an oblong block *k*, and in this the matrices for receiving the dies are placed.

The machine derives its motion from a shaft carrying the fast and loose pulleys E and F, driven by the belt *b*; this shaft carries a fly-wheel H, for imparting a uniform velocity to the working parts of the machine. To give the requisite power, a pinion G is fixed upon this shaft, and works into a wheel I, keyed upon another and stronger shaft situated directly over the former, and at each revolution of the wheel I, the machine performs one stroke. The ratio of the pinion G, to the wheel I, is as 1 to 6; consequently, when the pulleys are driven at the rate of 42 revolutions per minute, the machine performs 7 strokes per minute, which is found to be the most suitable velocity.

On the axis of the wheel I, is fixed a cam *c*, of the form denoted by the dotted lines in fig. 1. This cam, in its revolution, alternately raises and suffers to fall by its own weight, the friction pulley *d*, which runs loose upon the centre pivot of a *knee joint* composed of the arms *e*, *e*, and *f*, *f*. The arms *e*, *e*, working upon a fixed centre, as shown in the plan, the elevation of the pulley *d*, by the cam *c*, necessarily impresses a horizontal motion upon the corresponding extremities of the arms *f*, *f*. These extremities are connected by a joint to the slide *g*, the motion of which is guided into a strictly rectilinear and horizontal direction, by the dove-tail pieces *h*, *h*, planed true and screwed firmly to the frame of the machine. The sliding piece *g*, is furnished at its outer extremity with three holes or matrices for receiving the die *i*, which forms the outer head of the rivet. These matrices are so placed, that their centres coincide exactly, both in the horizontal and vertical planes, with the centres of similar ones in the upper extremity of the stem or riveting block *c*, already described. Into these latter is fitted the die *j*, against which the interior head of the rivet is placed during the process. The centre matrix in which the dies are represented in our drawing, is used for riveting every description of flat or circular work, while those at each side are required for finishing the corners of boilers. Thus the machine is adapted for riveting vessels of almost every shape within the given depth.

Action of the Machine.—The plates to be riveted together having been previously punched in the usual way, are suspended by a block and chain, as shown in fig. 1. The heated rivet is then inserted into its appropriate hole, and the attendant workman shifts the plates, so that the head of the rivet falls into the recess on the point of the die *j*. The machine is then put in motion, by changing the position of the strap *b*, from the loose to the fixed pulley. This motion is transmitted by the mechanism above described, to the sliding tool-holder *g*, and its projecting die *i*, in its advance, forms the head and finishes the rivet. The velocity of the machine is so calculated as to allow time between each stroke for the insertion of another rivet, and the re-adjustment of the plates, and thus the work proceeds without interruption.

It is stated by Mr. Fairbairn, that, with two men and two boys attending to the plates and rivets, his machine can fix, in the firmest manner, eight rivets of three quarters of an inch diameter in a minute, whereas, by the common process of hand riveting, three men and a boy can only rivet up forty per hour. Thus the quantity of work done in the same time in the two cases is in the proportion of 480 to 40, or as 12 to 1, exclusive of the saving of one man's labour.

Literal References.

A, A, the sole or base of the machine.
B, B, the side framing, containing bearings for the working parts
C, the upright stem or riveting block.
D, a malleable iron strap passing round the sole.
a, projections for receiving the ends of the strap D.
b, the driving belt or strap.
E, the loose pulley.
F, the fixed driving pulley.
G, pinion on the pulley shaft.
H, the fly-wheel for maintaining a uniform velocity.
I, spur-wheel on the cam shaft.
c, the cam by which the stroke is made.
d, a friction wheel in the centre of the knee-joint *e*, *e*, *f*, *f*.
e, *e*, *f*, *f*, the arms of the knee-joint.
g, the slide containing the matrices for the riveting dies,
h, *h*, fixed guides of the slide *g*.
i, *j*, the riveting dies.
k, a rectangular block forming the upper extremity of the stem C.
l, a small hand wheel and screw for fixing the movable die.

PLATE LII.

STEAM PUNCHING AND RIVETING MACHINE, BY M. LEMAITRE, PARIS.

The principle on which the motive power of the steam-engine is applied in the machine now before us, is widely different from that which characterizes most of those of which we have yet treated, and, simple and obvious as it may appear, it is only beginning to be appreciated by mechanicians as we think it deserves.

In those machines in which a rectilinear motion, whether in a horizontal or vertical direction, is required to be produced, that object has hitherto, in most cases, been accomplished by means of mechanism, more or less complicated and expensive, for converting the rotatory motion transmitted through long trains of shafts from the fly-wheel of a steam-engine, into a rectilinear motion. In establishments in which a great number of machines, small as well as great, have to be kept in action, we believe that no improvement upon this circuitous mode of transmitting the motion could be recommended; but for single and independent machines, where great power acting in a rectilinear direction is required, we believe that the *direct* action of the steam engine, as exemplified in the machine now to be described, may be employed with advantage. We have already given an example of the application of this principle, to a certain extent, in the Deal Sawing Machine, Plate XXVII.; and we may mention Mr. Nasmyth's direct action steam forge-hammer and pile-driving machine as additional eminently successful examples, which will be familiar to most of our readers.*

Another very important peculiarity in the machine now under consideration deserves special notice. In riveting by hand, the workman finds it necessary to bring the plates upon which he is operating into close contact, by striking them with his hammer, while closing and finishing the head of the rivet. The necessity of this will be obvious when we consider that the iron pin, which is to form the rivet, tends, by the compression to which it is subjected by the blows of the hammer, to *stave up* throughout its whole length, as well as at the end, and that, consequently, unless the plates are brought into very close contact during the operation, an obstacle to their perfect junction is interposed by the very means employed to bring them into intimate contact. In M. Lemaitre's machine this difficulty is obviated by a very ingenious and effective contrivance, which we shall now proceed to describe.

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* We have given detailed drawings and descriptions of Mr. Nasmyth's highly ingenious direct acting machines in the Supplement to this Work.

Fig. 3 is an elevation. Fig. 4, a plan. Fig. 5, an end view, and fig. 6, a partial section of the steam punching and riveting machine.

General Description.

The plates to be operated upon are, in this machine, placed horizontally between the fixed and movable dies *a*, and *b*. The matrix of the fixed die *a*, is at the extremity of a strong malleable iron stem or riveting block A, fixed firmly into the sole and foundation of the machine, and serving as the point of resistance against the action of the punch *l*, the compressing ferrule *i*, and the riveting die *b*. This last, which, as well as its corresponding fixed die *a*, is made of hard tempered steel, is fixed into a malleable iron stock or tool-holder B, accurately planed and adjusted to slide, in a vertical direction and without lateral motion, in a socket G, the farther purpose of which will hereafter be described. The tool-holder B has an alternate rectilinear motion of ascent and descent communicated to it by a wrought-iron lever C, which has its centre of oscillation at the upper extremity of a strong frame D, cast in a piece with the sole or base by which the machine is fixed to its foundations.

The opposite end of the lever C is connected by the rod E, to a piston working in the cylinder F. This cylinder, which is open above, and close beneath the piston, is furnished with a valve enclosed in the valve-box *c*; by this valve high-pressure steam is alternately admitted under the piston, through the steam-pipe *g*, and allowed to escape through the exhaust-pipe *h*. The valve is raised or depressed by hand, by means of the combination of rods and levers, *d*, *e*, *f*, which are disposed so as to place the machine within the command of the workman who superintends the operation.

The mechanism by which the plates are compressed during the process of riveting, consists of a cylindrical steel ferrule *i*, fig. 6, through the centre of which the riveting die *b*, passes, and which again is fitted into a strong cast iron socket G, sliding exactly and without play between two planed guides H, H. The socket G is made hollow for the purpose of forming the guide to the tool-holder B, as we have already mentioned, and receives a motion similar to, though independent of, the latter, from the two wrought-iron levers I, I, which have their centre of oscillation at the same point as the lever C, and are connected at their opposite extremities by the rod J, to a piston contained within the cylinder K. This latter cylinder is of smaller diameter than that used for riveting, and like it, is provided with a valve c', for the admission and escape of steam. The rods and levers d', e', f', for opening and shutting this valve, are arranged in the same way as those already described.

This machine is adapted for punching as well as riveting iron plates. For this purpose two strong parallel guides M, M are fixed to the movable frame which carries the compressing ferrule *i*. To the centre of these guides a socket or tool-holder L is attached by means of a pin *m*, passing through its upper extremity and the guides. The punch *l*, is fitted to the opposite end of the socket L, and its position, when brought into operation by turning it downwards upon its pivot *m*, and securing it between the guides M, M, coincides exactly with that of the matrix a', upon the extremity of the riveting block A. The matrix a' is sunk into a circular recess cast upon the riveting block, and for the sake of accurate adjustment, is acted upon by three small screws passing through the sides of the recess, as shewn in fig. 5. Under these circumstances, it is obvious that the operation of punching will be performed by the same mechanism, by which, in riveting, the compressing ferrule is made to descend. When not required to be used the punch and its socket are turned into the position represented in figs. 5 and 6.

Action of the Machine.—The plates to be united by riveting, having been previously punched, are placed together, as shown in the drawings, upon the horizontal stem or riveting block A. The heated rivet is then placed into its appropriate hole, with the head inside, and the plates are shifted till the rivet-head falls into the matrix *a*. The attendant workman by pulling down the handle f', depresses the valve enclosed in the steam-chest c', and thus opens a communication through the pipe g', between the steam in the boiler, and the under side of the piston working in the cylinder K. The piston ascends, and its motion being communicated through the rod J to the levers I, I, causes the ferrule *i*, to descend, and compress the plates firmly together. The same workman then, by pulling down the handle *f*, opens the valve of the large cylinder F, taking care that the pressure is still kept upon the plates. This causes the tool-holder B, and riveting die *b*, to descend, and thus the rivet is finished. The valve of the cylinder F is then first moved so as to shut the communication between the boiler and the piston, and allow the steam to escape through the pipe *h*. The weight of the rod E and lever C causes the piston to descend and the die *b*, to rise. The handle f', is then released, and by a similar process the ferrule *i*, rises to admit of the plates being shifted for the fixing of the next rivet.

The action of the machine in punching is obviously so similar to that already explained as to require no farther description. As it is of importance that the rivet-holes should be pierced with as little delay as possible, at the same distance from each other, and in the same line, M. Lemaitre makes use of a marker which serves as a guide to the workmen in placing the plates under the action of the machine. This contrivance consists of a small arm *n*, (fig. 5) formed into a socket, so as to admit of being adjusted and fixed upon the axis *o*, which has its bearings in one of the cheeks or guides M. Into the arm *n*, is fixed a small piece of sheet-iron, shaped at the outer extremity into a circle, the diameter of which is equal to that of the punch, and its centre, when turned towards the punch, coinciding with it. In making use of this contrivance the handle *p*, of the axis *o*, is turned round till the extremity of the arm *n*, is brought directly under the punch. The plates are then shifted so that the place where the hole is to be pierced is covered by the circular end of the sheet-iron marker, and thus the accuracy of the work is ensured.

Mechanism for Riveting Tubes.—Figs. 7 and 8 represent a very ingenious and most useful contrivance which M. Lemaitre has adapted to this machine, for the purpose of riveting long and narrow tubes from the interior, a problem which, at first sight, is of very difficult solution. The stem or riveting block N is made hollow throughout its whole length, and encloses a long rod S terminated by a steel wedge *r*. This wedge acts upon the movable matrix *t*, which passes through the upper side of the riveting block N, in such a way as to cause it to rise or fall according as the rod S is pushed in or drawn out. The die *u*, and its holder or stock O, are of the same form, and are connected with the machine in the same manner as those already described. In using this form of the machine, the rivets are inserted from the outside of the tube, the dies *t*, and *u*, receive simultaneously a motion in opposite directions, the lower one *t* being made to rise by pushing in the rod S, and the upper one *u*, descending by the action of the steam-piston upon the lever, and thus the rivet is formed.

Literal References.

A, the wrought-iron riveting block.
a, matrix or die on the end of the riveting block.
a′, matrix of the punching tool.
B, the stock or tool holder into which is fixed
b, the riveting die.
C, lever for conveying motion to the riveting die.
D, the frame and sole of the machine.
E, the connecting rod between the lever C and
F, the steam cylinder in which the power required for riveting is generated.
c, the valve box of the steam cylinder F.
c′, a similar valve box attached to the cylinder K.
d, *e*, *f*, rods and levers for working the valve of the cylinder F.
d′, *e*′, *f*′, similar rods and levers for working the valve of the cylinder K.
g, *h*, steam and exhaust pipes of the cylinder F.
g′, *h*′, steam and exhaust pipes of the cylinder K.
G, a socket or tool holder carrying the compressing ferrule *i*, and moving between the planed guides
H, H, bolted to the frame of the machine.
I, I, lever for conveying motion to the compressing ferrule.
J, the connecting rod between the lever I and
K, the steam cylinder in which the power required for compressing the plates is generated.
L, the socket into which is fitted
l, the punching tool.
m, a joint by which the punch-holder L may be turned upwards or downwards as required.
M, M, guides for confining the tool holder L laterally.
k, a strengthening piece by which the foundations, frame, and riveting block are held together.
n, a small sheet iron arm or marker.
o, axis upon which this arm is fixed.
p, a handle by which it may be turned out or in as required.
N, the hollow riveting block used for riveting tubes internally.
O, the stock or die holder used in the same operation.
S, a long rod having at its extremity
r, a steel wedge moving within the riveting block N.
t, the internal riveting die acted on by the wedge *r*.
u, the external riveting die set in motion by the action of the steam.

PLATES LIII., LIV.

PLATE-BENDING MACHINE, BY ROBERT NAPIER, GLASGOW.

This species of machine, originally confined to the tinsmith's shop, has recently,—enlarged in its dimensions, and rendered more complete in its mechanism,—become indispensable in the operations of boiler-making and iron ship building, in which plates are required to be bent to various degrees of curvature. The example given is the design of Mr. Elder, the manager of Mr. Napier's works, and is one of the largest yet made, being intended principally for use in the building yard, where plates of greater thickness come under operation than those required in boiler-making.

Fig. 1 is a side elevation of the machine; and

Fig. 2, an end view towards the driving geer.

Fig. 3 is a plan corresponding to fig. 1.

Fig. 4 is a vertical section in the plane of one of the end standards of the machine, showing the relative positions of the main centres, and the arrangement of the adjusting apparatus of the upper and oblique rollers.

Fig. 5 represents one of the rollers detached.

Figs. 6, 7, 8 and 9 are transverse sections of the framing, taken at different points, and fig. 10 is a vertical section through the rollers, shewing the mode in which the machine acts.

General Description.

The frame of the machine consists of two very strong end standards A, A, cast with soles to admit of their being bolted down to a solid stone foundation. They are braced together by four strong rods of malleable iron *a*, *a*, *a*, the screwed ends of which pass through projections on the standards, of such thickness that the nuts by which they are secured are nearly flush with the outside of the frames, thus obviating the necessity for having the geering far overhung.

The three rollers B, C, D are solid; the dimensions of each being twelve inches diameter, and ten feet long within their bearings. The two rollers B and C are placed in the same vertical plane; but the third roller D moves in a plane inclined at an angle of thirty degrees to the vertical plane. On one end of the roller B is keyed a strong pinion E, which geers with another pinion of the same size marked G, on the end of the lower roller C. On the opposite end of this last is fixed a large spur wheel H, into which works a pinion I, on the shaft J, which has a bearing in each of the two standards, and carries on its other end a large spur wheel K, commanded by a pinion upon the driving shaft L. One end of this shaft is carried in a bearing in the adjacent standard of the machine, and the other in an independent standard M, bolted to the foundation. It carries four pulleys, two of which N, O are fast upon the shaft, and the other two, P, Q, of double the breadth of the former, are loose. On these are two belts, the one cross and the other open, so that the rollers may be driven in either direction according as the motion is communicated by the open or the cross belt. The belts are shifted on their pulleys by means of the hand-bar R, on which are the guide-arms S, T, so placed that in reversing the motion of the machine, the belt thrown out of action shall have passed entirely from its fast pulley before the other shall have passed from its loose pulley to the fast one. This arrangement obviates the injurious effect so common in machines furnished with this species of driving geer, of the one belt acting against the other during a part of the time of shifting, thereby occasioning much unnecessary tear and wear of the belts.

The upper roller B is adjusted to the thickness of the plate to be bent by two strong set-screws U, U, which work in hexagonal brass nuts inserted into recesses in the standards, as shown in figs. 4 and 6. These screws bear against a steel plate resting upon the brass block in which the roller revolves, and which is of course placed over the journal, the pressure being upwards; and thus the roller is kept pressed down upon the plate, as it passes through the machine in the operation of bending. The mode of fitting the brass bearings of this roller, and also those of the oblique roller D, into the framing, is shown in fig. 7.

The roller D is adjusted to the required position by the double hand-crank on the upper end of the vertical spindle *b*, *b*, which communicates by means of a pair of small bevil wheels *c*, *c*, with the screws *d*, *d*, working into long brass nuts *e*, *e*, inserted in recesses of the frame. The upper ends of these nuts support the bearings of the roller, which may consequently be raised and lowered at pleasure, according as the spindle is turned in one direction or the other. The nuts *e*, *e*, are prevented from turning round with the screws by feathers upon the back of the brasses fitting into grooves on the ends of the nuts.

In machines of this kind it is common to provide geering for working both ends of the shifting roller simultaneously, so that the rollers may always preserve their parallelism. But in the present example that mechanism has been purposely omitted, to adapt it for bending the same plates with different degrees of curvature at the opposite edges; a description of work much required in the building yard, and which can only be effected by raising one end of the moveable roller D proportionally higher than the other.

Action of the Machine.—From this outline of the arrangement, the action of the machine will be easily understood. Motion being communicated through one of the belts to the driving shaft L, the pinion upon the end of that shaft geering with the large wheel K, gives motion to the shaft J; this motion, by means of the pinion I, is transferred to the large wheel H, which is fast on the under roller C. This roller being put in motion, communicates an equal velocity to the upper roller B, by means of the pinions E and G, which geer together. The roller D, the position of which determines the degree of curvature of the plate under operation, is only driven by the friction of the plate against it as it passes between the two other rollers.

Literal References.

A, A, the end standards of the machine.
Fig. 6, horizontal section of the standard on the line *f* — *g*.
Fig. 7, section on the line *h* — *i*.
Fig. 8, section on the line *j* — *k*.
Fig. 9, inclined section of standard on the line *l* — *m*.
a, *a*, *a*, stay rods, 2 inches diameter, for connecting the two standards.
B, the upper roller, 12 inches in diameter, and 10 feet long between its bearings, which are 5½ inches in diameter.
C, the lower roller, of the same dimensions.
D, the shifting roller, of the same dimensions.
b, the upright spindle for setting the roller D.
c, small bevil pair worked by the spindle *b*.
d, a screw, 2⅝ inches diam. and ⅝ in. pitch.
e, brass nut into which the screw *d*, works.
E, a pinion on the end of the top roller B; pitch 2½ inches, number of teeth 16.
G, a similar pinion on the lower roller C.
H, a large spur wheel on the opposite end of the roller C; pitch 1¾ inch, number of teeth 100.
I, a pinion of 13 teeth, working into the spur wheel H.
J, a wrought-iron shaft conveying motion to the pinion I from
K, a spur wheel at the opposite end of the machine; pitch 1½ inch, number of teeth 75.
L, the driving shaft of the machine.
M, a standard for supporting the exterior end of the driving shaft.
N, O, P, Q, fast and loose pulleys for setting in motion, stopping and reversing the machine.
R, the hand bar for shifting the belts.
S, T, guide arms fixed on the hand bar R.
U, U, screws for adjusting the top roller.

PLATE LV.

SELF-ACTING COMPOUND PLANING MACHINE, BY NASMYTH, GASKELL AND CO., PATRICROFT, NEAR MANCHESTER.

The machine represented in this Plate is remarkable for compactness and elegance of arrangement, and for the accuracy and dispatch with which a description of work that, previously to the introduction of such machines, could only be entrusted to the most expert and skilful mechanic, can, by its means, be executed by workmen of the most ordinary capacity. It is especially applicable to the finishing of the numerous small levers used in locomotive engine and tool-making, and is admirably adapted, not only to the planing of the sides and edges of such levers, but also to the finishing of their rounded ends, which otherwise could only be accomplished by the rude and tedious process of chipping and filing.

Fig. 1 is a side elevation of the machine.

Fig. 2 is a view of the front end or face.

Fig. 3, a general plan, and

Fig. 4, a transverse section through the principal working parts.

General Description.

The frame upon which the machine rests, and which is used for the purpose of raising it to a convenient height, is composed of two cast-iron cheeks A, A, strengthened by flanges, and held together at the lower end by two stay rods *a*, *a*. These frames are disposed at an angle to each other, in order to give greater stability to the structure. The main body of the machine consists of a cast-iron table or box B, bolted to the frames by internal flanges, as shown in the section fig. 4, and on the upper side of this table are cast the bracket C, carrying the driving spindle, and the rectangular chamber D, furnished with bearings for the other working parts of the machine. The square cast-iron sliding bar E, which carries the tool-holder, is accurately planed, and fitted into a recess in the upper portion of the piece D. It is of essential importance that the slide E should move accurately and without play in a rectilinear and horizontal direction, and for this purpose it is secured laterally by the adjusting screws *b*, *b*, and vertically by the wrought-iron plate, or cover *c*, fixed to the frame by the six countersink screws *d*, *d*, *d*.

On the front end of the square slide E is cast a flat rectangular plate, which serves as the fixed point of resistance to the various motions of which the tool-box is susceptible. The first of these is a rotatory motion, which is impressed upon it by a toothed quadrant plate *e*, worked by an endless screw on the axis *f*. This arrangement enables the workman to set the tool at any required angle to the work. On the upper edge of that part of the tool-box marked F, is fixed a nut through which works a screw *g*, surmounted by a handle or small hand-wheel. This screw is used for raising or depressing the tool, and thereby adapting it to the diameter or height of the work to be executed, as well as for regulating the depth of cut. The part G, which is thus acted upon by the screw *g*, is furnished with two parallel cheeks accurately dressed on their internal surfaces, and fitted to receive the tool-holder *h*. The tool itself is inserted into a square hole passing through the piece *h*, and fixed firmly to it by two pinching screws. The tool-holder *h*, is so formed as to admit of a small amount of rotatory motion round two centre screws passing through the cheeks of the piece G, and by this means accidents arising from the friction of the tool against the work in the return stroke, are prevented.

The mode in which motion is communicated to the tool-box is as follows: The extremity of the square slide E opposite to that on which the tool-box is fixed, is traversed by a longitudinal slot or groove *i*, adapted for the reception of a bolt-head, as shown in the transverse section, fig. 7. The projecting part of the bolt passes through a hole in the end of the connecting rod H, the opposite end of which is attached by another bolt to a rectangular cast-iron piece I, fixed to the end of the driving spindle, and acting as a crank for converting the rotatory into a rectilinear motion. The crank I is traversed throughout its whole length by a slot *i*, the form of which, as well as of that in the slide E, is shown in the section, fig. 7. By means of these slots the length of the stroke and the position of the tool may be easily and accurately adjusted to suit the work, as will be sufficiently obvious by inspection of fig. 1. The driving spindle works in two bearings, one of which, as before mentioned, is cast on the bed of the machine B, and the other is formed in the extremity of a bracket bolted to the side of it. The velocity of the driving spindle is varied and regulated by means of the cone-pulley J and fly-wheel K.

The planing of circular surfaces is effected, in this machine, by means of a hollow cylindrical cast-iron mandril L, (figs. 4 and 5), accurately turned and fitted into the body of the machine, the centre being exactly under that of the square slide E. This mandril is provided with a conical bearing on the front end, and is traversed by a malleable iron bolt *l*, secured to its opposite extremity by a nut. The head of the bolt *l*, is formed into a cylindrical socket, into which, by means of a cotter, is fixed another bolt, having two conical pieces *m*, *m*, one of which is immovable, and forms part of the bolt, while the other slides upon it, and is adjusted by means of the nut *n*. These pieces are used for the purpose of fixing the work M, upon which the machine is to operate; and from their conical form, adapt themselves to any required diameter, so as to ensure, without any trouble in setting, the concentricity of the outer surface with the eye.

The motion of the mandril L, with its appendages, is effected by means of the worm-wheel N, which is fixed to its inner end by a large circular nut screwed to the cast-iron mandril itself, independently of the bolt *l*, which passes through it. The worm-wheel N geers with an endless screw on the horizontal axis *o*, working in two bearings, one of which is formed by a small malleable iron piece bolted to the body of the machine D, and the other extends considerably beyond the table, and is supported by the bracket O bolted to it, as shown in the section, fig. 4. The axis *o*, besides the self-acting mechanism which we are about to describe, is provided with a handle, by which it can at pleasure be moved by the attendant workman. The self-acting geer, which, in fig. 6, is shown detached from the machine, consists of an eccentric P, fixed upon the driving spindle, and by means of the rod *p*, *p*, communicating a reciprocating motion to the slotted lever Q, which motion is again conveyed by the rod *r*, *r*, to the smaller slotted lever S. The centres of motion of these levers are respectively on the extremities of the traverse screw *v*, which is used only in flat planing, and on that of the axis *o*, of the endless screw, and on these centres they are fitted to move loosely without turning them. The slots which traverse the levers are for the purpose of altering the feed to any required amount. The motion of the levers is communicated to their axes by the double pauls *q*, and *s*, which work respectively into the wheels R and T, fixed upon their centres *v*, and *o*. These wheels act in this case as ratchet-wheels, and from the peculiar form given to the pauls, they may be made to move either backwards or forwards by simply reversing the direction of the pauls.

For the purpose of planing flat surfaces with this machine, it is provided with a cast-iron face-plate U, U, which is traversed by several slots, cast on its exterior surface, and adapted to receive the bolts by which the work is to be fixed to it. The back of the face-plate is planed and fitted to move transversely along the slide V, by means of adjustable dove-tail pieces, in the manner we have already so frequently had occasion to describe. This motion is effected by means of the screw *v*, which is worked by the mechanism above described, and which, having a bearing at each end of the slide V, passes along a recess cast in its surface, and works into a brass nut fixed on the back of the face-plate U. The traverse screw *v*, is provided with a handle *w*, for the purpose of setting the work into its proper position under the tool, before bringing the self-acting mechanism into geer.

This machine may be also used for nut-cutting, for which purpose, the cast-iron bearing W is bolted to the upper edge of the face-plate U, and through it passes the bolt *x*, on which the nut to be operated upon is fixed. A circular disc X is fixed to the opposite end of the bolt *x*, and on the circumference of the disc are several notches corresponding with the faces of the nut to be finished. The handle *y*, and catch and spring to which it is attached, are employed for the purpose of retaining and shifting the faces of the nut, in the same manner as we have described in treating of the common nut-cutting machine, Plate XLII. The nut being fixed on the bolt *x*, and adjusted by means of the catch, with one of its faces uppermost, is brought under the action of the machine by turning the handle *w*, of the traverse screw, and the operation proceeds as in ordinary flat planing. A flask Y is placed upon a standard screwed into the cover of the square slide E, and contains a solution of soap and water for the purpose of lubricating the work while under the action of the tool.

Action of the Machine.—It has been before remarked that this machine is peculiarly applicable to the finishing of the rounded ends of levers, and, in describing its action, we shall, in the first place, suppose that this is the description of work upon which it is to be employed. The eye of the lever, having been previously bored and faced in the turning lathe, or other boring machine, is fixed to the mandril L, in the manner shown in the detail fig. 5. The connecting rod H is next adjusted so as to give the amount of stroke necessary, in order that the tool may traverse the whole depth of the eye. The paul *s*, being disengaged from the teeth of the wheel T, the handle of the axis *o*, is then turned, till, by the action of the endless screw upon the worm-wheel N, the mandril L and the work which is fixed to it, are brought into the position proper for commencing the operation. The paul *s*, being set into geer with the wheel T, to the right or left according to the direction in which the work is to proceed, and the paul *q*, being disengaged, the machine is then set in motion. The crank I converts the rotatory motion of the driving spindle into a rectilinear motion on the sliding bar E, which carries the cutting-tool, and, at each revolution of the driving spindle, the eccentric P, which is fixed upon it, by its action upon the rods and levers *p*, Q, *r*, S, causes the wheel T and axis *o*, to which it is fixed, to revolve through an arc proportioned to the distance from their respective centres, at which the ends of the rods *p*, and *r*, are fixed to the slotted levers Q and S. The mandril L is thus made to revolve, and, at each stroke of the machine, to bring a fresh portion of the work under the action of the tool. By lessening the amount of feed on the mandril, and of cut on the tool, levers may thus be finished so smoothly and accurately as not to require the application of the file.

For the purpose of planing flat surfaces, that portion of the mandril L which is inclosed within and projects in front of it, is removed by simply unscrewing the nut at its inner extremity, and withdrawing the bolt *l*, and the parts attached to it. The work is then fixed to the face-plate U by bolts passing through the slots on its surface, and being brought under the action of the tool by turning the handle *w*, of the traverse screw *v*, the paul *q*, is then set into geer with the wheel R, and the rod *r*, is removed. The machine is then put in motion, and, by the action of the eccentric P upon the lever Q and traverse screw *v*, to which it is attached by means of the wheel R, and its paul *q*, the face-plate U is made to advance upon the slide V, in the same manner as in the common planing machine.

Literal References.

A, A, the framing or supports of the machine.
a, *a*, stay rods connecting the frames.
B, B, the table or sole of the machine.
C, bracket cast upon the table, for supporting the driving spindle.
D, a rectangular mass cast upon the upper surface of the table, with recesses for the working parts.
E, square sliding bar carrying the tool box.
b, *b*, pinching screws for adjusting the slide E.
c, the wrought-iron cover of do.
d, *d*, *d*, screws for fixing the cover *c*.
e, quadrant plate for setting the tool to any required angle.
f, axis and handle for working the quadrant plate.
g, screw for setting the tool vertically.
F, back part of the tool box.
G, front part of do.
h, the tool holder.
H, connecting rod between the slide E and
I, the cast-iron crank on the extremity of the driving spindle.
i, *i*, slots for adjusting the length of stroke and position of the tool.
J, the driving cone pulley.
K, fly-wheel on the driving shaft.
L, the hollow mandril by which the work is turned on its axis.
l, bolt passing through the mandril L.
M, the work to be operated upon.
m, a bolt, with conical pieces for fixing the work.
n, the nut of the bolt *m*.
N, the worm wheel by which the mandril is turned by the feed motion.
O, bracket for supporting
o, an axis, with endless screw working into the worm wheel N.
P, the eccentric upon the driving spindle.
p, *p*, the eccentric rod connected to
Q, a slotted lever on the traverse screw *v*.
q, the paul on the slotted lever Q.
R, spur wheel acting as a ratchet for turning the traverse screw *v*.
r, a rod connecting the lever Q with
S, the slotted lever on the axis *o*.
s, the paul by which the motion of the lever S is imparted to
T, a double ratchet fixed on the axis *o*.
U, the face-plate, with slots for fixing the work.
u, an adjustable dove-tail piece fitting to
V, the slide on the front of the machine, used in flat planing.
v, the traverse screw of the face-plate V.
w, handle of the traverse screw.
W, a bearing for the bolt used in nut cutting.
X, circular disc, with notches for retaining the faces of the nuts.
y, a catch, with handle and spring for setting the disc X.
Y, flask with lubricating material.

PLATES LVI.—LXIII.

MARINE STEAM-ENGINE OF ONE HUNDRED AND FORTY-FIVE HORSE POWER, BY CAIRD AND CO., GREENOCK.

These Plates illustrate very fully the form and construction of marine engines made by Messrs. Caird and Co. of Greenock, for the steam-packets Actæon and Achilles, and also for the royal mail packet Urgent, still plying betwixt Liverpool and Dublin. The drawings were made from the engines of the Actæon, since lost on the West India station; but, in order to render them more complete, and more acceptable to the engineer, the expansion geer subsequently applied to the engines of the Achilles has been introduced. It may also be remarked that by proportionally reducing the scale of the drawings, they will be found to agree in every respect, beyond a few very slight modifications of a technical kind, with the larger class of engines, (severally of 225 horse-power), since constructed by the same spirited firm for the West India mail-packets, Clyde, Tay, Tweed and Teviot. The Plates may thus be regarded as giving a general representation of the form of marine engines built by a firm to whose engineering skill the profession is indebted for a design equally remarkable for elegance of appearance and compactness of arrangement. In lightness of material it is no doubt surpassed by the recent introduction of malleable-iron framing, and direct action; but, in the class to which it belongs, known as *side-lever engines*, it exhibits a massiveness of appearance, and an economy of weight which, in combination with equal strength, have not hitherto been surpassed.

Enumeration of the Plates.

Plates LVI., LVII., contain a complete side elevation of the engine, showing the general design and arrangement of the framings, and the relative positions and connections of the working parts; the valve, expansion, and starting geer, parallel motions, and situation of the pumps. In this view the side of the vessel is supposed to be removed and the engine seen in its proper position.

Plates LVIII., LIX. Fig. 1 is a plan of the sole-plate of the engine with all the superincumbent parts removed, and showing the position and mode of attachment of the steam-cylinder bottom and valve casing, the hot-well, placed on the top of the condenser, the air-pump, and the soles of the main framing.

Fig. 2 is a general plan of the engine, exhibiting very distinctly the starting and eccentric geer, the mode of working the pumps, the direction and position of the steam-pipe, and mode of connecting the diagonal framing; also the relative horizontal positions of the valve and expansion geer.

Plates LX., LXI. This sheet contains a general section of the engine, corresponding to the elevation above mentioned. This view exhibits the internal arrangements, and the mode of fitting together the more important parts, such as the steam-cylinder and piston, valve casing and valves, condenser and air-pump, foot and discharge valves, &c.

Plates LXII., LXIII. This sheet is occupied with views of parts which could not be fully shown in combination. The leading dimensions of the more important of these parts are marked in figures in these Plates.

Fig. 1 is a face view of the steam-cylinder looking on the steam-ports.

Fig. 2 is a plan of the cylinder and nozzles with the covers of both removed.

Fig. 3 combines both a face and back view of the valve casing, showing on the left the exterior form of the packing ports with their covers removed; and on the right the interior form of the same parts.

Fig. 4 is a face view of the steam-valves, exhibiting the manner in which the two valves are connected together.

Fig. 5 represents the main centre and also the form of cross section of one of the side levers.

Fig. 6 is a side view of the piston cross-head; and fig. 7 is a corresponding view of the cross-tail and link.

Fig. 8. A side view of the air pump cross-head, and a section of the feed-pump, of which fig. 9 is an elevation, and fig. 10 a plan.

Figs. 11 and 12 shew the parallel motion rods detached.

Figs. 13 and 14. The throttle valve and its spindle.

Fig. 15 is a side elevation, and fig. 16, a corresponding section of the feed chest; fig. 17, a face view; fig. 18, a back view; fig. 19, a plan of the top, and fig. 20, a plan of the bottom.

Figs. 21, 22, 23, and 24 are various views of the injection valve casing; fig. 25 shows the injection valve itself in elevation, and fig. 26, in section.

General Description.

The Sole-plate and Condenser.—The sole-plate marked A, A, with the condenser U, consists of a single casting, double ribbed on the under side to give it additional strength and rigidity. For facility of fitting, it is furnished with fitting strips on its upper surface; these are dressed true to receive the soles of the main frame and cylinder bottom, which are fitted upon it metal to metal, being provided with corresponding fitting strips. The sole-plate is firmly secured to the keelsons of the vessel by sixteen strong malleable iron bolts marked *a*, *a*, *a*, in the elevation, and the recesses for which are similarly designated in the plan of the sole-plate. The middle of the plate, between the two keelsons, is depressed to the level of the under edge of the strengthening ribs, in order to allow the condenser and its appendages to stand lower in the vessel than they otherwise would, as shown in the general section.

The Framing.—The main framing B, B, of the engine consists of four strong fluted columns, cast pair and pair with their soles and entablatures. The soles, as above observed, are fitted upon the sole-plate metal to metal; and are secured to it by bolts and dove-tail keys, for which strong projecting seats are cast on the sole-plate. The entablature is completed by two cross beams corresponding in form with the sides into which they are fitted, and secured by bolts. The form of their cross section is shewn in the general section of the engine.

The upper or crank framing, also marked B, B, consists likewise of four columns cast pair and pair with their soles and entablatures; but in this case there are no cross pieces, the two sides being simply braced together by two strong malleable iron stays *c*, *c*, running transversely through and between the framings of both engines. One of these passes between two strong projecting pieces cast on the back columns near the top, and the other between the cheeks of the *diagonal framing* C, C. The crank framing rests on the entablature of the main frame to which it is fitted, and secured by bolts and by two centre keys in either sole, driven on each side of a dove-tail projecting piece cast on the entablature of the lower frame, and which enters a similarly formed but larger recess in the sole of the upper frame—an arrangement which is clearly shown both in the general elevation and section. This framing is farther secured between the ship's beams by the strong stays *b*, *b*, cast upon the entablatures. These stays usually abut against cast-iron face-plates, bolted upon the paddle-beams at the points of contact, but they are neither bolted nor otherwise fixed to the face-plates, being left free to slide vertically upon them in obedience to any spring which the vessel may have when under way, and which is often considerable, especially in a rough sea. The crank framing is also braced to the steam-cylinder by the diagonal framing C, C, consisting of two strong parallel struts cast upon the inner columns of the crank framing, from which they spring. These struts terminate in rectangular flanges, abutting against similarly formed projections cast on the top of the cylinder, on opposite sides of the valve casing; and to these they are carefully fitted metal to metal, and secured by bolts, as partially shown at *d*, in the elevation and plan.

The principal use of the crank framing, and that from which it takes its name, is to support the crank-shaft. This is accomplished in each of the engines by two plummer-blocks, one on each side of the crank, secured by bolts and keys on the entablatures of the frames. The soles of the plummer-blocks are in this, as in all highly finished engines of the same class, likewise faced and fitted metal to metal.

The Steam Cylinder.—The steam-cylinder E is cast open and with broad and strong flanges, at both ends. It is placed upon a separate bottom piece, flanged like the cylinder, to allow of their being bolted together. This bottom piece being truly faced on its upper and under surfaces, is secured to the sole-plate of the engine by strong bolts, and the interstices of the joint filled with iron cement. The lower end of the cylinder is similarly fitted upon and secured to the upper flange of this bottom piece, so that the whole may be perfectly steam-tight.

The interior of the cylinder is bored as truly as possible of a uniform internal diameter of sixty-two inches. The cover *d*, which, as will be observed from the section, is cast hollow, in order to prevent the loss of heat by radiation, is fitted into the upper end to nearly the depth of the steam-port. On the inside is a circular recess to receive the heads of the bolts of the piston junk-ring; and the exterior plate is expanded by a strong flange to the same diameter as the flange of the cylinder, to which it is secured by numerous bolts. In the centre is formed the stuffing-box through which the piston rod ascends.

The external form of the cylinder is most distinctly shown in the separate views given of it in Plate LXII., figs. 1 and 2. The projecting corner pieces marked *d*, are those to which the diagonal framing is attached; they are faced both on their horizontal and vertical surfaces, so that the corresponding flanges, in which the struts of the framing terminate, may be fitted truly upon them. In fig. 1 is also shown the form of the projecting faces against which the slide valve works, and which are cast of a piece with the cylinder. These are carefully dressed, and the whole surface of both exactly reduced to the same plane; and to complete them a carefully finished facing of brass is applied steam-tight round each of the ports, and is provided with projecting tails also carefully finished, flush with the general bearing surface, and of a length proportioned to that of the ledges of the valves. The outlines of these faces are indicated in the view above referred to, and the transverse form in the general section of the engine.

The Piston.—The body of the piston *a*′, (see general section), consists of a single hollow casting, strengthened by radiating feathers, with a strong eye in the centre to receive the piston rod. The under side is a portion of a sphere coinciding with the curvature of the bottom of the cylinder. The upper side in like manner is convex to its junction with the ring *b*′, *b*′, which is flat, and fits to a corresponding part on the upper surface of the piston. By making these parts of a curvilinear section, they are better secured from rupture by changes of temperature; and the piston being symmetrical in its outlines with the cover and bottom, the loss of steam due to the *clearance* is reduced to a minimum.

The lower edge of the piston only is cast of a diameter equal to that of the cylinder, the deficiency on the upper surface being made up by the ring *b*′, *b*′, called the *junk-ring.* This ring is fitted steam-tight, first by turning, and subsequently by grinding. The packing rings *c*′, *c*′, consisting of two thicknesses, made up of overlapping segments, are likewise fitted into their place between the junk-ring and the flange corresponding on the under side of the piston, and are rendered steam-tight by the same means. The whole of these thicknesses, composing the edge of the piston, are simultaneously turned of a uniform diameter, precisely equal to the internal diameter of the cylinder, in which they are intended to work steam-tight; but as this condition would endure only for a short period, however carefully and exactly the fitting might be effected, were no provision made for compensating for the wear incident to the continued motion of the piston, (and especially under variations of temperature, as in starting), the packing rings are rendered capable of adjusting their diameter to that of the cylinder, by springs placed behind them and abutting against the parts *a*′, *a*′. The springs employed in this piston are of a U-form, placed vertically, the strong side bearing upon the piston at *a*′, and the elastic side against the adjacent ends of two of the segments. The number of springs and segments is thus necessarily equal, and they are so arranged that every segment is acted upon at each end by a spring. By this means the piston is made to work in the cylinder steam-tight, and to accommodate itself to any slight variations due to the contraction and expansion of the materials; and likewise to compensate for wear of its own circumference and that of the cylinder.

The junk-ring *b*′, is secured in its place by bolts and nuts; the nuts are placed in recesses provided for them in the metal of the piston, and the bolts are screwed into them from the outside. The heads thus project on the surface of the ring, and would come into contact, at the end of the up stroke, with the cylinder cover, but for the circular recess formed in it for their reception, as before noticed.

The piston rod H is inserted into the piston through the central eye, which is bored, and tapered from the under side. The rod is of malleable iron, and turned, the body of it truly cylindrical, so that it may work freely in the stuffing-box through which it ascends, and the lower end with the same taper as the eye into which it is fitted, and in which it is secured by a cotter traversing the metal of the eye and the thickness of the rod, and thereby effectually binding the latter to the piston.

The Piston Cross-head and Connections.—The piston rod, ascending through the packed stuffing-box e', is inserted into an eye, bored a little smaller than its own diameter in the cross-head I, equi-distant from the two extremities; and is then fixed by two gibs and a cotter in the usual manner. The cross-head, shown detached at fig. 6, has two journals turned on each of its ends, separated from each other by narrow collars. By means of these journals the radius bars K, K and the side-rods J, J are connected with the cross-head. The side-rods are fitted on the exterior journals, and descend to the corresponding extremities of the side-levers M, M, to which they are also flexibly attached. Their connection at the cross-head, as will be observed from the elevation, is by solid eyes formed in the ends of the rods and bushed, the brasses being retained in their places by the collars of the cross-head, with the assistance of a cotter bearing against the back of the lower brass; but the connection with the side-levers is effected differently. Here the brasses are placed within strong malleable iron straps, bent at the middle of their length till their projecting ends fit closely upon the rectangular ends of the side-rods, to which they are secured by a gib and cotter. This species of connection, technically known as the *butt and strap*, and universally adopted in similar circumstances, provides for any slight wear of the brasses; for should these become too large, they can be brought closer together by driving the cotter more tightly, the holes through which the gib and cotter pass being so disposed in the strap and butt that the gib shall only be in contact with the ears of the strap, and the cotter only with the lower edge of its hole in the butt. The holes in both strap and butt being thus of greater breadth than the gib and cotter together, the connection admits of adjustment to the extent of this difference, and no farther, for then the edges of the holes being in the same plane, the relative positions of the strap and butt will not be altered by any subsequent action upon the cotter.

The side-levers are forked at their extremities at the points of connection with the side-rods J, J, and the links *j*, *j*, which connect them with the cross-tail O. The joints are completed by strong malleable iron pins which pass through the jaws of the levers, and the bushes of the straps which are placed between them. These centre pins are turned with a little taper on the parts which pass through the levers, and the holes made for their reception are accurately bored to the same diameter and ground. The studs are driven in tightly from the outside, and secured in their places by riveting at the opposite extremity.

Both levers are suspended upon the same axis M, called the *main-centre*, which passes through and is fixed in the sides of the condenser. It is shown detached in fig. 5, together with the form of cross section of one of the side-levers in the plane of the axis. From this figure it will be observed that the eye of the lever is fitted with brasses which can be tightened as they wear by a pair of cotter-keys, passing through the boss *i*, and bearing against the back of the lower brass. Inside, and bearing against the shoulder of the boss, is a ring of malleable iron, of sufficient breadth to cover the margin of the eye to fully an inch beyond the circumference of the brasses, thereby preventing the lever from deviating inwards; and to prevent it from shifting its position outwardly, a plate of the same external diameter is applied by a strong bolt screwed into the end of the main-centre.

The side-levers are jointed to the connecting rod N by means of the wrought-iron cross-tail O and links *j*, *j*. The cross-tail, with one of the links, is represented in fig. 7. The connecting rod passes through an eye at the middle of its length, and is fixed by two gibs and a cotter, in the same way as the piston rod is attached to the cross-head, while the links are connected in the same manner as the side-rods, except that the upper ends do not admit of adjustment, being simply riveted in the eyes. The attachment of the connecting rod with the crank P is also effected by means of a butt and strap, the cotter of which is tightened and maintained in its place by a screw and nuts *k*, as shown in the general section. The crank-shaft Q rests, as before observed, on pedestals fixed upon the entablature of the crank framing, and is prevented from longitudinal displacement, by collars formed upon it and fitted between the flanges of the brasses.

It may be noticed that the piston rod, side-rods, cross-head, main-centre shaft, cross-tail and links, connecting rod, and crank shaft are all formed of the best malleable iron, and are carefully turned and dressed to the requisite forms and dimensions.

The parallelism of the piston rod is maintained during the stroke of the engine by the two radius bars K, K, one of which is shown separately at fig. 11; by the radius levers *f*, *f*, fast upon the ends of the shaft L, called the *parallel motion shaft*, and by the parallel bar *h*. The ends of the radius bars K, K, which are attached to the cross-head, are formed with solid eyes, fitted with brasses, and tightened by cotters, in the same manner as those of the side-rods which are attached at the same points. The eyes at the opposite ends, which work upon studs in the radius levers *f*, *f*, are formed in the same way, but are smaller, and have the outer brasses adjustable by screwed pins *g*, *g*. The length of the bar thus admits of slight adjustment between its centres, to compensate for errors of workmanship, and wear of the bushes. The parallel bar *h*, shown detached at fig. 12, is also connected by a solid eye and stud to the lever *f*, and admits also of adjustment to a considerable extent at its lower end. This bar, it will be observed, is formed in two pieces with the contiguous ends screwed right and left, and embraced by a long nut similarly screwed. By turning this nut in either direction, it is obvious that the upper and lower ends will be made to approach or recede, and the distance between the centres be thereby diminished or increased. Its lower end is jointed to the exterior side-lever by a wrought-iron stud inserted into a rectangular eye, and secured by a nut.

The disposition of the centres of motion in this arrangement is such, that in every position of the piston the angles of the parallelogram formed by the part of the side-lever comprehended between the stud of the rod *h*, and the junction of the side-rod, opposed to the radius bar K, and by the parallel bar *h*, and the side-rod J, shall change proportionally, always preserving the same constant ratio; it follows therefore that the piston rod cross-head will move constantly in the same plane, and the parallelism of the piston be thereby maintained agreeably to the principles laid down in our general notice of Parallel Motions, (see Plates XIII., XIV).

The parallel motion shaft L is supported by two plummer-blocks, resting on the entablatures of the small pillar-framing D, surmounting the diagonal framing. It consists of four columns, cast, like the larger framings, pair and pair, with their soles and entablatures, and with provision on the latter for bolting and keying the pedestals. The soles are necessarily oblique to correspond with the upper flange of the diagonal framing C, C, to which they are secured by bolts and keys.

The Valves and Valve-geer.—The valve casing F, which is cast of a semi-cylindrical form, corresponding to the form of the valves, is fitted steam-tight and bolted to the side of the cylinder by projecting flanges cast on both for that purpose; and also to the sole-plate over the recess T', shown in the general elevation. The flat side, as will be observed from the general section, occupies only about a third of the whole length equi-distant from both ends, and is cast with projecting flanges, which are carefully fitted steam-tight between the inner surfaces of the projecting faces of the cylinder. By this construction it will be observed that when the cover m', is applied, all communication between the interior of the valve casing and external atmosphere is shut off; the passages T and T' open into the condenser U, and the steam-ports f', f', into the cylinder.

The steam-pipe G, G likewise opens into the valve-casing, and a communication is thereby effected with the boilers. To prevent the pipe being injured by expansion arising from the variations of temperature to which it is liable, it is provided with *expansion joints*, which allow the ends to slide upon each other, and thereby maintain the same aggregate length between the two extremities. It has also a valve, called the *throttle-valve*, placed in it to regulate the supply of steam, and to cut off the communication between the boilers and the casing when necessary. This valve is placed close to the junction of the pipe with the casing, as shown in fig. 3; it is simply a disc fitting to the inside of the pipe, with a hexagonal eye cast in it to receive the spindle upon which it works. Two views of the valve itself are given in fig. 13, and two views of the spindle in fig. 14.

The steam valve h', h', consists of two short D-slides firmly connected together, one to each port of the cylinder. The backs are turned to a true semi-cylindrical surface, and the faces are planed and ground to the brass faces of the ports, so that they may slide upon them steam-tight, and with as little friction as possible. They are kept tight against the faces, and also rendered steam-tight in the casing by hemp packings, introduced through the *packing-ports e, e*, cast in the casing. These packings are covered by the packing rings k', k', which are pressed against them by set-screws acting in nuts between the packing rings and the port covers, as fully shown in the general section. The set-screws may be tightened at pleasure by a box key, inserted through holes formed in the covers, and fitted with hollow plugs, which can be withdrawn when necessary.

The planes forming the faces of the valves are slightly less than double the breadth of the port, but the circular parts are necessarily much larger. The faces and backs are connected by strong diaphragms, through which pass the ends of the rods i', i', which couple the two valves together, as shown both in fig. 4, and in the general section. These rods are turned to an exact length between the collars against which the contiguous sides of the diaphragms bear, and are retained in their places by nuts upon their protruding ends. To increase their rigidity, they are strengthened at the middle of their length by the cross-stay j'. A strong stud l', is inserted downwards in the middle of the diaphragm of the upper slide, and is secured in its place by a nut on the end projecting below. The upper end is formed with oblong cheeks or jaws between which the flattened end of the valve-spindle is received, and retained by a strong square pin passing through the cheeks of the stud and the end of the spindle. The hole in the spindle is made slightly oblong to allow the valve, as it becomes worn by the friction of the faces, to be pressed forward by the packing without straining the spindle.

The valve-spindle, passing through a packed stuffing-box in the cover m', of the valve casing, is attached by means of a small cross-head and side links, to the lever *n*. This lever is fast upon the transverse shaft *m*, which has its bearings immediately under those of the parallel motion-shaft in the framing D. On the opposite end of the lever *n*, is fixed a weight *q*, sufficiently heavy to counterpoise the weight of the valves. This weight is connected with the shaft S, called the *starting shaft*, by two small levers *s*, *s*, attached to the axis of the weight *q*, by two short connecting rods jointed to each. The shaft S is carried on pedestals fixed upon the lower flange of the diagonal framing C, C, and has a short lever keyed upon the end projecting to the *inside* of the engine. A long bar called the *starting bar*, is fitted to this lever, but, being required only occasionally, it is made capable of being removed, and for that reason is not shown in the drawings; but supposing it applied, it is plain that by moving it towards the right, the shaft S will be carried with it in the same direction, the balance weight *q*, will be elevated, and the valves depressed. The reverse action will produce the reverse effect by again lowering the weight and raising the valves. Now, observing in the section that the lower steam-port of the cylinder is open to communication with the condenser by the passage T', and that the upper port communicates only with the interior of the casing, if the weight *q*, be raised until the valves descend through a space equal to the breadth of the faces, it is clear that the conditions will be reversed; the upper port will then be opened to communication with the condenser through the passage T, the passage T' will be shut, and the lower port will open into the interior of the casing.

Upon one end of the traverse shaft *m*, is keyed a lever upon the stud of which a gab formed in the end of the compound rod *l*, *l*, called the *eccentric rod*, rests. This rod, which consists of two slight wrought-iron bars rendered rigid by diagonal braces, is attached at its opposite extremity to the brass ring of the eccentric R, which revolves with the crank-shaft of the engine; consequently, supposing the crank to revolve, the rod l, l, will at the same time receive an alternating rectilineal motion, which being transferred to the crank lever of the traverse shaft m, will cause the ends of the lever n, alternately to ascend and descend. But the valve spindle being attached to this lever, its motion will be transferred to the valves, which will thus be made alternately to ascend and descend in the same manner as when a lever is applied by manual strength to the shaft S. This is the action necessary to maintain the motion of the engine, as will presently be explained.

The parallelism of the valve spindle is maintained by means of the links o, o, arranged as in the common parallel motion of stationary engines. The radius rods are attached to opposite sides of a small framing consisting of two columns, fixed on the cover of the valve casing, and having their entablature p, of a semi-circular form, to allow the cotters of the cross-head links to pass clear of it.

The Condenser and its Appendages.—The condenser U, and the lower exhaust passage T′, T′ are cast of a piece with the sole-plate. Two strong bosses are formed in the sides of the condenser to receive the main-centre shaft M′, which passes completely through it. In the top is an opening for the upper exhaust passage T, T, the vertical part of which is cast of a piece with the hot-well X, though separated from it by a partition, as shown in the general section.

The cold water for effecting the condensation of the steam which passes into the condenser by the exhaust ports, is admitted by the injection pipe n'. This pipe passes through the side of the vessel, and communicates with the water without; but, in order to regulate the supply, a valve is placed in the pipe, close upon the condenser; its position is marked by v, in the plan of the sole-plate, and its construction is fully exhibited by the various projections given at figs. 21—26, as already enumerated. The face upon which the slide works is formed on the side of the condenser at v, over which the casing is fixed. The mode of working the valve is by a small brass spindle which rises through a packed stuffing-box in the cover of the casing; this is attached to a long lever passing to the opposite side of the engine, and which can be more or less depressed at pleasure, to allow of a larger or smaller supply of injection. The part of the pipe within the condenser, passes completely across, and is perforated with numerous small holes to diffuse the water more completely in the body of the condenser, and thereby render it more effective. A shelf o', is fixed over the orifice of the lower exhaust passage T′, in order to prevent the water from falling within it, and at the same time to diffuse it more effectually in the condenser.

The Air-Pump and Valves.—The lower part of the condenser communicates by a valve p', called the *foot-valve*, with the air-pump V, the barrel of which projects within the sole-plate to a short distance from the bottom of the passage, leaving space for a body of water to enter it from below. The barrel of the pump is bored and lined with a thin cylinder of brass. A strong flange is cast around it near the lower end, for the purpose of bolting it water-tight to the margin of a circular opening cast in the upper division of the sole-plate for its reception, as shown in the general section, and also in the plan of the sole-plate.

The bucket, which is formed of brass, consists of a ring connected by four arms q', q', to the central eye, into which the rod is fitted. The under side of the ring has a flange cast upon it, between which and the projecting ledge of the junk-ring, bolted on its upper surface, a packing of hemp is retained. The pump rod is sheathed with brass, to prevent corrosion by contact with the water. To apply this sheathing, the rod, which is of malleable iron, is first roughly turned; it is then thoroughly cleaned and taken to the brass foundry, where the covering of brass is cast upon it, of somewhat more than the required thickness. It is again put into the lathe and turned to the requisite diameter. The rod, which thus possesses all the advantages of strength and diminished liability to corrosion, is retained in the tapered eye of the bucket by a cotter, and passes through a packed stuffing-box in the cover t', of the pump. The bucket valve r', is of that kind technically known as the *pot-lid valve*, in contradistinction to the *butter-fly valve*, which consists of two hinged flaps. The pot-lid valve is a circular brass plate capable of sliding vertically on the pump rod by means of a bored eye at its centre; the plate is strengthened by ribs radiating from the eye, and terminating in a narrow ring on its circumference, which is faced and fitted water-tight by grinding, upon the similarly prepared edge of a ring projecting round the plane of the bucket.

To understand the action of this valve, it is only necessary to conceive the under part of the barrel to be filled with water, and the bucket to be forced to descend in it, the foot-valve p', being shut, to prevent its return into the condenser; it is then obvious that the water passing between the arms q', q', will meet the under surface of the valve r', and prevent it descending with the bucket, thus forcing a passage at its circumference, over the ring of the bucket; and will continue to ascend *relatively* in the barrel so long as the bucket continues to descend. But when the bucket has attained the lowest point of the stroke and begins to return, the valve, being of greater specific gravity than the water, will shut by its own weight, and will carry whatever water is above it, to the height of its own ascent. The water thus elevated is ejected by the valve s', called the discharge valve, into the *hot-well* X, so called because the water thus thrown into it by the air-pump, being that employed in condensation, has its temperature considerably increased.

The foot and discharge valves p', and s', together with the spindles which form their axes, are cast of brass. The box-framings of these valves are formed of cast-iron, faced with brass, and fitted water-tight into their seats, where they are each retained by two long copper keys, one at each side, inserted from above before the covers are put on. The covers are likewise fitted water-tight and bolted down. The valves are prevented from opening beyond the requisite distance, by projecting bridges situated before them, as shown in the general section.

The Hot-well.—The hot-well, as already observed, is situated above the condenser. The part marked X, with the vertical part of the upper exhaust passage T, is formed of a single casting, fitted water-tight to the top of the condenser. In one side of the well is a rectangular recess covered by a door, through which admission can be obtained into the interior; and in the side adjacent to that of the vessel, as shown in the elevation, is a circular opening to which the discharge pipe Y is bolted. Through this the excess of water, beyond that required for supplying the boilers, is discharged into the sea. The pipe consists of a single length outside of the condenser, to which it is fitted by an expansion joint, to compensate for the spring of the vessel when at sea; and has also a valve in it capable of opening outwardly, but which being shut, resists the pressure of the water inwards.

An air-vessel Z is placed over the hot-well and fitted to it air and water-tight. The object of this vessel is to create a *head pressure* by means of the column of water which it is capable of containing, to assist in ejecting the water through the discharge valve should the hot-well from any cause become surcharged. The pressure thus brought into action by continuing to increase with the exigency of the case will, under all ordinary circumstances, prevent accumulation of water in the well to any detrimental extent; and for further security, a small discharge pipe is attached to its upper extremity.

The Feed and Bilge Pumps.—The same cross-head u, by which the air pump is worked, serves also to work two other pumps of smaller dimensions. These are the feed pump, by which water is supplied from the hot-well to the boilers, and the bilge pump, by which leakage water is withdrawn from the hold of the vessel. The cross-head and a section of the feed pump, which is very nearly identical in construction with the bilge pump, are shown separately in fig. 8. Fig. 9 is an external elevation of the same pump. The barrel is formed of cast-iron, but the plunger, which is made hollow for the sake of lightness, is of brass. It is connected to the cross-head by a cotter, a portion of the end being made solid for that purpose.

This pump communicates with the hot-well by a pipe projecting from the side of the barrel, a portion of which is shown in the separate views, and also in the general elevation, where it is marked W. The feed-box, shown separately in figs. 15—20, is bolted upon the part of the hot-well formed in the condenser, (at the position marked w' in the plan of the sole-plate), by square flanges upon the face, as seen in fig. 17. In the side of the hot-well are two square holes corresponding to the two openings in the feed-box, and these being made to coincide, the pipe from the feed pump is attached to the lower of the two circular openings, (fig. 18), in front of the box, while the upper communicates by a copper pipe with the boilers, a clack valve being placed in each of the two lower divisions of the feed-box. Now, supposing that the plunger is made to ascend in the barrel of the pump, leaving a corresponding space unoccupied, the water will flow from the hot-well into the pump through the lower of the square holes; but on the plunger beginning to descend, the valve in that division of the feed-box will be closed by the pressure of the water tending to return to the hot-well, and consequently will be forced through the upper valve, and along the feed pipe, to the boilers. But if more water be drawn by the pump than is required for the boilers, it is simply ejected through a loaded conical valve in the upper division of the box, and thus returned to the well. The weight on this valve can be adjusted at pleasure according to the pressure of steam in the boilers.

Instead of being guided by parallel motion bars, the pump cross-head is restricted in its vertical path by two fixed guide rods v, v, passing through bushed eyes in the cross-head, which slides freely upon the rods in its alternating ascent and descent. The cross-head is connected to the side levers by the rods t, t, which are formed with solid bushed eyes at their upper ends, and with butts and straps at their lower extremities.

The Snifting Valve.—The bottom of the air-pump well communicates by a pipe, as shown in the general section, with a small conical valve marked u', and which is technically called the *snifting valve.* This valve is kept shut by a screwed pin v', passing through a wrought-iron bridge made fast upon the mouth of the pipe. To the side of this pipe, above the valve, is cast a small return branch, by which the water passing through the valve is carried off. The use of this valve is to admit of the escape of the air within the condenser, air-pump and steam passages, on starting the engine, and before these have been filled with steam. When about to start, the pin v', is simply unscrewed by hand, to permit the valve to rise so that the air and water may escape, and give place to the steam, which now flows onwards from the valve casing, occupying all the passages and condenser, and finally begins to issue by the valve u'.

The Blow-through Valve.—This valve is situated at the position marked u in the plan of the sole-plate. It is placed in a chest fixed upon the steam valve casing at the lower end, and has two openings, one above and the other below the packing port. The valve itself is placed between these two apertures. This valve is used simultaneously with the snifting valve, to admit steam into the passages and condenser, when preparing to start the engine, and thereby to displace the air and water which may be lodged in them, through the snifting valve.

The Priming Valves.—These are two small valves g', g', situated in the steam-ports of the cylinder, as shown in the general section, and are called *priming valves* from their being intended to discharge any water carried over into the cylinder with the steam, and which is technically termed *priming.* These valves are kept shut by springs acting against them externally, and of such strength as to resist the ordinary pressure of the steam; but should water lodge in the passages, owing to its non-elastic properties, it will be ejected through the valves by the action of the piston tending to compress it.

The Expansion Geer.—The expansion geer consists of an apparatus by which the amount of steam admitted during a stroke of the piston can be diminished at pleasure, when it is not required to work the engines to full power. The first part of the apparatus consists of a cam with five faces fixed on the crank-shaft, as shown in the elevation and plan of the engine. These faces are of different lengths, giving five different degrees of expansive action. They are so formed that the friction roller on the end of the lever *w*, against which the cam acts, is thrown forward through the same space; but the time of action varying with the length of the face, the effect will depend upon the particular face in contact with the roller; and this, according to its distance from the frame, may be made to bear against either one or other of the faces. The position of the roller, and consequently of the lever to which it is attached, is regulated by a screw and nut; the latter is formed in the back lever *x*, which is forged of a piece with the weighted lever, and has a long hollow boss working on a stud fixed in the framing. On the projecting end of the screw is fitted a handle, which being turned, causes the lever *w*, to advance or recede upon the boss of the double lever, on which it is guided longitudinally by a sunk key. The roller is kept against the face of the cam by the action of the weighted lever; the weight tending to descend and carry the lever with it, causes the opposite lever to press upwards against the face of the cam.

The lever *x*, is connected by a joint with an adjustable rod, carried forward to the lever *y*, which is fast upon a cross shaft supported by two small columns on the flange of the expansion chest, marked G in the elevation. On the same shaft is keyed the double-ended lever y', one end of which is connected by intermediate links, with the spindle of the expansion valve, which is of the kind known by the name of *equilibrium* valves. The opposite end of the lever y', communicates by a rod *z*, with an arrangement of levers attached to the side of the condenser, by which the apparatus can be thrown into geer and disengaged at pleasure. Thus the end of the crank lever z', being moved to the right, the rod *z*, will be drawn down, and with it the end of the horizontal lever y', to which it is attached; but the lever y', being fast upon the same axis as the vertical lever *y*, this latter will be thrown back, and at the same time the lever *x*, with which it communicates; consequently the lever *w*, will be projected forward, and the roller thrown out of contact with the cam, when the engine will receive the full supply of steam.

Action of the Engine.—To bring the engine into action, the steam is allowed free admission into the valve-casing, by the steam-pipe G, G, leading from the steam-chest over the boilers. The steam-ports of the cylinder being both shut by the valves, and the blow-through and snifting valves open, the steam is allowed to pass into the valve casing by opening the throttle-valve, partially at first, which fills the steam-passages and condenser, driving the air and water before it. When this has been accomplished, and steam alone issues by the snifting valve, the blow-through valve is closed, and the injection valve is opened; the cold water now rushing into the condenser, effects the condensation of the steam with which it was filled, and creates the desired vacuum. The eccentric rod *l*, being out of geer with the crank upon the traverse shaft *m*, and a long lever, as before described, being applied to the starting shaft S, the steam valves are raised until the under port communicates, by the passage T', T', with the condenser, and the upper port with the interior of the valve casing, now full of steam, which, in consequence of this disposition, will flow into the cylinder above the piston, and force it to descend. The next operation is to draw the starting lever in the opposite direction, and thereby to reverse the position of the valves, shutting off the communication of the upper port with the steam in the casing, and opening it to the condenser, at the same time that the communication of the lower port is cut off from the condenser, and opened to the interior of the casing. This being done, the steam will flow from the cylinder into the condenser, and encountering there a shower of cold water from the injection pipe, will be condensed, and a vacuum thereby formed in the cylinder above the piston. By that means the pressure over the piston is removed, and the steam flowing into the cylinder beneath it, forces it to ascend to the top of the cylinder.

But the piston being connected, by the cross-head and side-rods, with the side-levers, carries these with it in its ascent and descent, through an arc, whose chord is equal to the length of the stroke of the piston; and the side-levers being connected at their opposite ends, by means of the cross-tail and connecting rod, with the crank, the motion of the piston is thus transferred to the crank-shaft, and through it to the paddles, which are fixed upon its outer extremities.

After two or three strokes of the piston the moving parts will have acquired a certain degree of momentum, and this is taken advantage of to render the engine self-acting. The crank shaft being in motion, if the eccentric rod *l*, be thrown into geer with the traverse shaft, exactly the same effect will be produced upon the valves as by the lever applied to the starting shaft S; for by the alternating thrust and pull of the rod, communicated to it by the eccentric R, the crank of the traverse shaft will be made to describe a certain portion of a revolution, proportional to the eccentricity of the eccentric, and the valve lever *n*, being fast upon that shaft, the valves must consequently ascend and descend regularly with the revolutions of the crank shaft; and these revolutions are performed uniformly with the alternating ascent and descent of the piston. The eccentric is not fixed immovably to the crank-shaft, but is carried round with it by means of catches, which are so adjusted as to adapt it for the reversing, as well as for the forward motion of the engine.

The water is drawn out of the condenser by means of the air-pump with the same regularity; for the air-pump cross-head being worked by the side-levers, it will move simultaneously with them; the feed-pump being also attached to the same cross-head, the boilers will be furnished with water in proportion to the speed of the engine, and consequently in proportion to the quantity of steam used.

Literal References.

A, A, the sole-plate of the engine.
a, *a*, *a*, holding-down bolts by which the sole-plate is fixed to the keelsons.
B, B, the crank framing.
b, *b*, spring-stays of the crank framing which work between face-plates on the paddle-beams.
C, C, the diagonal framing.
c, *c*, stay rods connecting the framings of both engines.
D, the parallel motion framing.
d, *d*, flanges by which the diagonal framing is bolted to the cylinder.
E, the steam cylinder.
a', the steam piston.

b', the junk-ring of the piston.
c', c', the packing rings of the piston.
d', the steam cylinder cover.
e', the gland or stuffing box-cover of the steam cylinder.
$f' f'$, the steam ports.
$g' g'$, the priming valves situated in the upper and lower steam ports.
F, the steam valve casing.
e, e, the packing ports on the back of the valve casing.
h', h', the steam valve, consisting of two short D-slide valves.
i', i', two wrought-iron rods connecting the valves h, h.
j', an intermediate rod for strengthening the rods i', i'.
k', k', the valve packing rings.
l', the stud for connecting the valve spindle to the upper D-valve.
m', the valve casing cover.
G, the steam pipe and expansion valve chest.
H, the steam piston rod.
I, the cylinder cross-head.
J, J, the cylinder side-rods.
K, K, the radius bars of the piston rod parallel motion.
f, f, the radius levers of the parallel motion.
g, g, pinching screws for adjusting the ends of the radius bars.
L, the parallel motion shaft.
h, the parallel motion side-rod attached to the lever f, and to
M, M, the great side-levers of the engine.
M′, the main-centre.
i, i, bosses at the centres of the side-levers, through which pass the keys for tightening the bearings.
N, the connecting rod.
O, the cross-tail of the connecting rod.
j, j, the cross-tail links.
k, gib and cotter with screw for adjusting the bearings of the upper end of the connecting rod.
P, P, the cranks.
Q, Q, the crank or paddle shaft.
R, the eccentric for working the valves.
l, the eccentric rod.
m, the traverse or valve shaft.
n, the valve lever.
o, small parallel motion for the valve spindle.
p, a small framing to which are attached the ends of the radius bars of the valve parallel motion.
q, the back-balance or counter-weight of the valve.
r, the back-balance or counter-weight of the eccentric.
s, s, levers by which the valve counter-weight is attached to
S, the starting shaft.
T and T′, the upper and lower exhaust passages.
U, the condenser, cast of a piece with the sole-plate.
n', the injection pipe.
o', the splash board for spreading the injection water.
p', the foot valve between the condenser and air pump.
V, the air-pump barrel.
q', the air-pump bucket.
r', the bucket valve.
s', the discharging valve between the air-pump and the hot well.
t', the air-pump cover.
u', the snifting valve.
v', screw and handle for opening and shutting the snifting valve.
t, t, the air-pump side rods.
u, the air-pump cross-head.
v, guides for the air-pump cross-head.
W, the feed pump.
X, the hot-well.
Y, the discharge pipe.
Z, the air vessel.
w, x, y, y', levers of the expansion geer.
z', rod and handle by which the expansion geer may be thrown into or out of action.

PLATES LXIV., LXV.

SELF-ACTING SLOTTING MACHINE. BY CAIRD AND CO., GREENOCK.

These Plates represent a machine adapted for slotting or paring work of moderate size, and for cutting the key-grooves or seats of wheels not exceeding five feet in diameter. It is at once elegant in design, simple in construction, and capable of adaptation to a great variety of circumstances.

Fig. 1 is a front, and fig. 2, a side elevation of the machine, shewing the general arrangement of the working parts. Fig. 3 is a general plan, and fig. 4, a transverse section of the work table and part of the framing on which it rests.

The framing consists of a strong fluted column A, with two brackets of proportionate strength for carrying the working geer and slotting bar, and a sole frame for supporting the work table and its appendages, and having a strong bottom plate by which the machine can be bolted to a stone foundation. The whole of this framing consists of a single casting, and therefore may be presumed to possess all the strength and rigidity which can possibly be obtained with the form adopted and the weight of metal employed; two conditions of the utmost importance in machines of this kind, in which the strain varies suddenly from the mere weight of the slotting bar to the maximum pressure necessary to effect the cut.

The projecting palms of the brackets are faced and formed with dove-tail edges, between which the slotting bar B slides in its up and down motion. Two of these dove-tail pieces are attached by screws, and can be adjusted by set pins, as they are worn by the sliding action of the bar. On the lower extremity of this bar the slotting tool is attached by two glands and set screws, in the usual way; and, at some distance from its upper end, it has an adjustable stud fitted into it, to which the upper end of the connecting rod O is jointed. The mode of fixing and adjusting the stud is clearly shown in the front elevation of the machine. From this view it will be observed that the bar has a long slot, occupying about a third of its length at the upper end, between the parallel cheeks of which the rectangular body part of the stud is accurately fitted. This part of the stud is formed with shoulders which bear against the inside of the bar, and has a strong screwed pin projecting from its exterior surface, on which a large pinching nut is passed. This nut being screwed tight against the face of the bar, the stud is effectually secured from shifting its position in the slot, by the friction induced between the bar and the shoulders of the stud on the inside, and the nut on the outside.

The rectangular body of the stud is traversed by a long square-threaded screw, which occupies the whole length of the slot in the bar, and which can be worked by the small hand-wheel P, fixed on its upper end. This screw is so fitted into the machine as to have no end-long motion independent of the bar; but when turned by means of the wheel P, on its upper extremity, it will cause the stud to assume any required position in the slot. But it is easy to perceive that by changing the position of the stud in the slot, the height of the slotting bar will be correspondingly changed in relation to the work table of the machine. In effect the stud may be considered as a fixed point by which the bar is suspended, and consequently by turning the hand-wheel of the screw in one direction or other, the bar will be correspondingly elevated or depressed, and the tool thereby set at any height above the table that may be necessary for the kind of work under operation. And when it is so adjusted, the stud is made fast in its place by tightening the pinching nut on the screwed tail projecting in front of the bar, as above described.

The lower end of the connecting rod O is flexibly attached by a stud-bolt to the disc or crank-wheel N, which has a radial dove-tail groove a, formed in its plane face to receive the correspondingly formed head of the bolt. This bolt or stud is embraced by a strong ferrule of slightly greater length than the eye of the connecting rod, which fits upon it freely; and being in its place, a large nut is passed upon the projecting end of the bolt, which fixes the ferrule between it and the edges of the groove in the face of the wheel, and thereby effectually secures the stud in the required position, while the connecting rod is left free to revolve on the ferrule, in consequence of the latter being slightly greater in length than the eye of the rod. The position of the stud in relation to the centre of the crank-wheel obviously determines the length of the stroke of the slotting bar. Thus, the wheel admits of the stud being fixed at seven and a half inches from the centre as a maximum, and therefore the utmost throw will be fifteen inches.

The crank-wheel N is fast upon the end of a strong shaft which passes through a long socket cast in the middle bracket of the frame, and which is bushed at the two extremities. These bushes are formed in halves, to admit of their being adjusted, as they wear, by cotters acting against the under brasses, as shown in the side elevation of the machine. On the opposite end of the crank-wheel shaft is keyed the spur wheel G, which geers with the pinion H, on the driving shaft. The cone-pulley K receives motion by a strap from the main shaft, and is susceptible of three modifications of speed, to suit the kind of work under operation, the fly-wheel I rendering the motion uniform, and obviating the jerks and variations to which it would otherwise be liable. This shaft has one of its bearings in the columnar frame of the machine, while the other is independently supported by a pillow resting on the sole of a wall recess.

The sliding table D is movable on the upper surface of the bed plate E, in a direction parallel to the sole frame of the machine; and the circular table C is capable of sliding horizontally on this last, in a direction at right angles to the direction of motion of the table D. From the sectional view, fig. 4, it will be observed that the table C is provided with a rectangular sole plate, to which it is attached by a central stud and socket, in such a manner as to be capable of working freely on the stud as on an axis. By this arrangement two motions of the upper table are obtained, one rectilineal and the other circular. The rectilineal motion is obtained from the sole plate, on which are bevilled ledges, adjusted to slide in corresponding faces formed on the table D, as shown in fig. 2; and the circular motion is obtained by causing the upper plate to revolve on its centre. The first of these motions is communicated by means of a screw *e*, which passes through a longitudinal recess, formed for its reception in the table D, and works into a nut attached to the sole plate of the upper table. To obtain the circular motion, the table C is formed with a worm wheel on its circumference, into which the worm on the spindle S geers; and as this spindle is attached by its bearings to the rectangular sole plate, which cannot revolve in consequence of its attachment to the table D, it is obvious that by turning the crank handle on the worm spindle, the plate C will be made to revolve on its central stud.

The table D can also be worked by hand, by placing a crank handle on the square end of the screw *h*, the self-acting mechanism to be described presently being out of geer. This screw has its bearings in the bed plate E, and works in a nut attached to the under side of the table. It may also be observed that one of the dove-tail or bevilled ledges of each of the sliding tables is adjustable by set screws when reduced by wearing, as shown in the section of the table D.

A self-acting motion may be given to the under table by means of an arrangement of parts shown in the side elevation of the machine. These consist of the ratchet wheel L, which is keyed upon the end of a spindle connected by a universal joint at M, with the screw *h;* and a pall *l*, attached to the end of the lever *m*, on the same axis and formed of a piece with the lever *n*. In one arm of the wheel G is fixed a stud *o*, carrying a small friction pulley, and adjustable, like the stud in the crank-wheel, to any required distance from the centre. This stud, as the wheel revolves, comes in contact with the lever *n*, which, being loose on its axis, yields to the pressure, and through the lower arm *m*, and pall *l*, transmits its motion to the ratchet wheel, and through this again to the screw *h*. The pall *l*, can be applied to either side of the ratchet wheel, so that the table may be made to travel upon the bed E, in either direction, and as the throw of the lever *n*, can be regulated by the position of the stud *o*, the amount of the feed motion may thus be adjusted to the kind of work. The object of the universal joint at M is to permit the table to be set at a small angle with the horizontal plane, when necessary, as in cutting the key-seats of wheels. This is effected by raising the inner end of the table by means of the screwed link *b*, jointed to the bed plate E, as shown in fig. 4, and acted upon by the set nuts marked 2, 2, shown in the side elevation of the machine. The bed plate of the table, when in this position, is supported by two palms fitted to a cylindrical piece formed on the front of the main sole; and is prevented from moving laterally by the set screws *g*, *g*.

The circular motion of the table C may be communicated by the handle on the end of the worm spindle S; but, to render it self-acting, a double ratchet wheel is substituted for the handle, and is worked by palls attached to a rocking lever which communicates by a series of small rods with the lever *m*. The transverse motion of the upper table can also be given by the handle on the end of the screw *e;* but a ratchet may also be substituted for this handle, and worked by a pall connected with the levers employed to transmit the circular motion of the table. Thus each and all of the three motions of the table may be rendered self-acting, and the work thereby carried on independently of that constant attention which would otherwise be requisite on the part of the workman. It is seldom, however, that more than one of the self-acting motions is required to be in action at a time, the other motions being adjusted by hand.

Literal References.

A, the frame of the machine.
B, the slotting bar.
a, dove-tail groove in the crank-wheel N.
C, the circular table upon which the work is fixed.
D, the under slide of the table.
E, the bed plate of this slide.
b, a link with adjusting nuts for setting the work table at an angle.
e, guide screw of the upper table.
g, *g*, set screws for preventing lateral motion of the table.

h, guide screws of the lower table.
G, spur wheel on the crank-wheel shaft, geering with
H, a pinion on the driving shaft.
I, fly-wheel on the driving shaft.
K, cone-pulley for driving the machine.
L, a ratchet wheel by which motion is transmitted to the slide screw *h*.
l, a pall for working the wheel L.
m, lower arm of the lever to which the pall is attached.
n, upper arm of the same lever receiving motion from the stud *o*.
M, a universal joint by which the spindle of the ratchet wheel L is connected to the screw *h*.
N, the crank-wheel on the main shaft.
O, the connecting rod to the slotting bar.
P, hand-wheel on the screw for adjusting the stroke of the machine.
S, the spindle of the worm geering with the worm wheel on the table C.

PLATES LXVI.—LXXIII.

LOCOMOTIVE ENGINE AND TENDER, BY ROBERT AND WILLIAM HAWTHORN, NEWCASTLE-UPON-TYNE.

In the history of that department of engineering to which these Plates refer, there is to be observed a striking dissimilarity to that of the Stationary Condensing Engine, the great monument of the genius of Watt. The latter came perfect, in every essential respect, from his hands, and all that succeeding mechanicians have been able to effect, in this particular department, has been to modify his arrangements, and to perfect the subordinate details of the engine. The case is widely different with the Locomotive Engine. From its first introduction it has gone through many progressive stages of improvement, and even now, so numerous and important are the amendments which are almost daily proposed and adopted, and so various are the forms and arrangements employed by the different makers, that it would be absurd to assert, either that any given specimen is a perfect one, or that it is undeniably superior to all others of the same date of production.

The example which we have chosen for detailed illustration, and which we are now about to describe, is the form of locomotive engine and tender at present constructed by Messrs. Hawthorn of Newcastle, a firm whose success and extensive employment in this branch of the trade is a sufficient guarantee for the excellence of their arrangements. In these Plates are embodied all the most recent improvements which Messrs. Hawthorn have introduced into their engine, including their patent auxiliary expansion frame and the mechanism by which it is moved.

It is our intention to describe the engine and tender, with their respective detailed parts, under distinct heads, so as to avoid confusion in the letters of reference.

The engine is made, according to the method generally adopted by Messrs. Hawthorn, with a cranked axle and outside bearings; it is furnished with six wheels, (designated a *six-wheeled engine*); the driving and fore wheels, which are five feet diameter, are coupled together, and the hind wheels, three feet diameter, are placed immediately below the fire-box. By this arrangement the greatest safety is ensured, and particularly at high speeds; the same amount of stability being given to the engine as if the hind wheels were placed *behind* the fire-box; with this additional advantage, that the length of coupling between the wheels may, by the present disposition, be regulated to any convenient distance. An engine of this description will be found exceedingly useful for general purposes, being adapted both for merchandise and for mixed or passenger trains at ordinary speeds; while for express or special trains, where a high rate of speed is required, railway travelling would be rendered comparatively safe, by employing engines specially made and adapted for such purposes.

Enumeration of the Plates.

Plates LXVI., LXVII., contain—

Fig. 1. A longitudinal elevation of the locomotive engine.

Fig. 2. A general plan of the same.

Fig. 3. A longitudinal elevation of the tender, showing the mode of its connection with the engine.

Fig. 4. A general plan of the tender, in which are seen the cocks for regulating the supply of water to the boiler, and the hand-wheel for working the brake apparatus.

Plates LXVIII., LXIX.—In this sheet are shown sectional views corresponding in their relative positions to the external views given in the last:

Fig. 5. A longitudinal section of the engine, showing the internal arrangements of the boiler, and the working parts of the engine.

Fig. 6. A sectional plan of the engine, with the cylindrical part of the boiler removed, for the purpose of exhibiting the general arrangement of the working parts, and the construction of the fire-box.

Fig. 7. A longitudinal section of the tender.

Fig. 8. A plan of the tender with the tank removed, showing the construction of the framing, drag springs, brake geer, &c.

Figs. 9, 10, 11. A plan, longitudinal section and transverse section of the slide valve and expansion slide frame, represented in their working positions. These figures are drawn to an enlarged scale of one and a half inch to the foot.

Plates LXX., LXXI.—This sheet is occupied with end elevations and transverse sections of the engine, and with enlarged views of Messrs. Hawthorn's patent expansion geering.

Fig. 12. An elevation of the engine as seen at the fire-box end.

Fig. 13. A transverse section through the fire-box.

Fig. 14. An elevation of the engine as seen at the smoke-box end.

Fig. 15. A transverse section through the smoke-box. In this view, the cylinder to the left is sectioned through the steam passage, while that to the right is supposed to be cut through the discharge port and blast pipe.

Fig. 16. An elevation of the geering for working the slide valve and expansion slide frame.

Fig. 17. A plan of the same.

Plates LXXII., LXXIII.—This sheet contains detailed views, drawn to an enlarged scale, of such parts of the engine and tender as could not be fully shown in combination.

Fig. 18. No. 1. A transverse section of the steam regulator and chest; No. 2, a longitudinal section of the same.

Fig. 19. No. 1. A plan of the piston, with the cover removed to shew the packing; No. 2, a section of the piston through the lines 1–2–3; No. 3, a plan of the same, complete, with the cover and guards.

Fig. 20. No. 1. A plan of the piston rod cross-head, with slide blocks and projecting arm for working the feed-pump; No. 2, a side view of the same; No. 3, an end view of the same.

Fig. 21. No. 1. An elevation of the backward eccentric; No. 2, a plan of the same, showing the stud for working the expansion geer.

Fig. 22. No. 1. A side view of the reversing or coupling link; No. 2, an edge view of the same, showing the stud by which the valve is shifted into forward or back geer.

Fig. 23. No. 1. An elevation of the front end of the eccentric rod; No. 2, a plan of the same.

Fig. 24. No. 1. A longitudinal section of the feed-pump, with the plunger, valves, &c.; No. 2, an end view of the same.

Fig. 25. No. 1. A plan of the double safety-valve, with the seat; No. 2, a longitudinal section of the same.

Fig. 26. No. 1. An edge view of the driving-wheel, half in section, to show the mode of fixing the arms, tyre, &c. In this view is also given part of the cranked-axle, to show the relative positions of the crank, wheel, bearing, &c.; No. 2, a side view of the driving-wheel.

Fig. 27. No. 1. A transverse section of the driving-wheel axle-box, and of part of the outer spring; No. 2, a longitudinal section of the same.

Fig. 28. No. 1. A section of the suspending link for adjusting the weight of the engine on the springs; No. 2, a side elevation of the same.

Fig. 29. No. 1. A general elevation of the tender brake geer; No. 2, a plan of the brake lever and toothed sector; No. 3, the screw and link-nut for the tender brake.

Description of the Engine.

The Fire-Box.—The first part of the engine which claims our attention is the fire-box. The form of fire-box which Messrs. Hawthorn have adopted is clearly shown in the end elevation, fig. 12, and transverse section, fig. 13. It consists of two parts: the external fire-box A, which in reality forms part of the boiler, being filled with water to about fifteen inches from the top; and the internal fire-box B, placed within the other, and which contains the fuel for generating steam. The internal fire-box is made of copper, and tapered slightly towards the top, for the purpose of allowing the globules of steam which are formed on its sides to ascend more freely. To resist the downward pressure of the steam, the roof is strengthened by the strong malleable-iron stays C, C, bolted across, and having a bearing against its sides, while both external and internal fire-boxes are secured against the lateral strain by having numerous iron stay-bolts *a*, *a*, *a*, screwed through both boxes, and riveted at each end. The fire-door *b*, affords access to the internal fire-box for the admission of coke. It is of an oval form, and the latch is provided with a chain for the greater convenience of opening and shutting. The space between the two fire-boxes at that part where the fire-door is situated, is filled by an oval-shaped frame, securely riveted through both, and the fire-door itself is furnished with a plate of iron riveted to the inside at some little distance from it, to save it from warping by the intensity of the heat within. The fire-bars *c*, *c*, distinctly shown in the section fig. 5, and in the plan fig. 6, are ranged parallel to each other on a wrought-iron frame fixed to the under side of the fire-box, and a portion of them, marked *d*, in the plan, is so arranged as to admit of their falling at one end on the removal of the pin which supports them. In this case the burning fuel drops into the ash-box D, fixed below to receive it, and the combustion almost immediately ceases.

The Boiler.—As before remarked, the external fire-box A forms part of the boiler, communicating freely with it, and being, like it, filled with water to the proper height when the engine is in operation. The boiler, properly so called, is marked E in our Plates, and in the specimen now under notice consists of a cylinder 11 feet 6 inches in length, and 3 feet 6⅝ inches in diameter outside. It is traversed throughout its length by 107 brass tubes *e*, *e*, *e*, 2⅛ inches outside diameter, of No. 13 and 14 wire gauge. These tubes are inserted into the front plate of the internal fire-box, (called the *tube-plate*), which is made of a sheet of copper considerably thicker than the other plates of which the fire-box is composed, so as to afford a better bearing for the fixing of the tubes. At the front extremity of the boiler, they pass through a similar plate of iron, which forms the partition between the boiler and the smoke-box. Into these plates the tubes are secured at both ends by riveting, and subsequently, by strong steel ferrules accurately turned, and driven firmly into the interior of the tubes, so as to render them perfectly tight and free from leakage. The cylindrical form of the boiler renders lateral staying unnecessary, and the tubes themselves, at that part where they are situated, secure it against the pressure in a longitudinal direction; but for further safety, three strong malleable iron stay-bolts *f*, *f*, *f*, traverse the whole length of the boiler, and are secured to it by round pins passing through brackets riveted to the front tube-plate, and to the back plate of the external fire-box. The whole boiler is covered externally with a coating of thick felt and with strips of wood, called the *lagging*, or *cleading*, to prevent the radiation of heat, as well as to give greater symmetry of appearance.

The Smoke-Box.—The tubes *e*, *e*, *e*, all open into that part of the boiler called the smoke-box F, the purpose of which is to collect the gases evolved by the combustion of the fuel, and to transmit them through the chimney G, into the air. In this compartment of the boiler are also placed the steam cylinders, and other very important parts of the engine, to be hereafter described. The front plate of the smoke-box is furnished with large folding doors *g*, *g*, fitted air-tight to it, and provided with a handle, by which both doors are simultaneously shut and opened. These doors, which are distinctly shown in the end elevation, fig. 14, and in the section, fig. 5, serve to afford access for the insertion and cleaning of the tubes, as well as for the examination and repair of the parts of the engine referred to above.

The Safety-Valves and Boiler Mountings.—Although the efficient working of the engine requires that the boiler be capable of generating steam of a high elastic force, it is yet essential to safety that the steam pressure be confined within certain limits. In order to ensure this, the boiler is provided with two safety valves, *h* and *i*, both placed in one chest, (see detail sheet, fig. 25,) fixed on the summit of the external fire-box, and surrounded by a polished brass chimney H, of a form symmetrical with that of the large chimney G. One of these valves, marked *i*, which is of the kind called the *lever safety-valve*, can be regulated to any required degree of pressure by the engine-driver, being furnished with a *spring-balance*, by which the amount of pressure is distinctly indicated. The other safety-valve *h*, is inaccessible and is loaded by a spiral spring and screws, to such a pressure as may be considered safe, yet higher than the engine is expected, under ordinary circumstances, to require.

To indicate the height at which the water stands in the boiler, and to enable the driver to keep it always at its proper level, a set of gauge-cocks and glass tube *j*, communicating with the water inside, are fixed at a convenient situation near the foot-plate. A graduated scale is fixed behind the glass tube, and the required level may thus be maintained with considerable accuracy.

As a precaution against accidents, and to give notice of the approach of the engine, a steam whistle *k*, is attached to the top of the fire-box, and communicates with the steam within by a short pipe, provided with a stop-cock. The internal construction of the whistle is such, that when the stop-cock is opened, the steam rushing out with great force encounters the sharp edges of a species of inverted cup, thereby emitting a shrill and very loud noise, which can be heard at the distance of several miles.

Behind and at the lowest extremity of the fire-box, is situated the *blow-off cock l*, by which the boiler may be emptied of water when required; and, for the purpose of cleansing it of the accumulation of sediment which is constantly being formed in it when the engine is in operation, it is provided with *mud-holes* both at fire-box and smoke-box ends. These mud-holes, which are shown in figs. 12 and 15, are secured when the engine is at work, by covers or *doors* bearing against the *inside* of the boiler, and fixed each by a single bolt passing through a strong wrought-iron bridge bearing against the outside.

The Steam-Pipes and Regulator-Valve.—The steam-chest or receiver I, rises from the centre of the cylindrical part of the boiler, and is carried to a considerable height above it, in order that the mouth of the steam-pipe J, which opens into it, may be removed to as great a height as can conveniently be obtained, from the surface of the water in the boiler. The object of thus raising the open orifice of the steam-pipe is to prevent *priming*, that is, the ascent of water along with the steam, and its consequent flow through the steam-pipe into the cylinders, where its presence in any considerable quantity would produce the most serious inconveniences, besides the danger to which the boiler would be exposed by its rapid abstraction. As a further precaution against priming, Messrs. Hawthorn make use of a simple but very ingenious contrivance. This consists of a species of inverted cone *m*, fig. 5, made of sheet iron, and riveted to the interior of the steam-chest, with an aperture in the centre just wide enough to allow the free ascent of the steam between it and the steam-pipe, which passes through it. The water in the boiler tends to prime chiefly where there is a surface of metal to which it may adhere; consequently, when, in rising up the sides of the steam-chest, it encounters the inverted cone *m*, its course is diverted downwards and towards the centre, where being unsupported it falls back into the boiler. Should any priming occur round the sides of the steam-pipe itself, the water is, in a somewhat analogous way, diverted by the bell-shaped mouth of the pipe, and returned into the boiler. The steam-receiver is surrounded by a polished brass dome, which, besides being highly ornamental to the engine, serves the very important purpose of diminishing the radiation of heat, by interposing a stratum of heated air between the steam-chest and the external atmosphere.

The steam-pipe J is made of copper, and that part of it which is enclosed within the boiler is 5¾ inches internal diameter. It enters an orifice accurately bored and fitted to receive it, in the cast-iron *regulator-valve chest* K, which is bolted steam-tight to the exterior of the front tube plate of the boiler. The valve-chest K incloses a regulator-valve *n*, of a new and improved form, which, as well as the chest itself, is shown on an enlarged scale in the detail sheet, fig. 18. It is formed of cast-iron, and has two projecting faces accurately and smoothly turned, and of such form and dimensions as, when placed in the position shown in fig. 18, No. 1, completely to cover the orifices of the two branch steam-pipes J, J, whose faces are bored truly cylindrical, and of the same diameter as that of the faces of the valve. The distance between the contiguous edges of the two branch pipes is somewhat greater than the breadth of the valve face, so that when turned round in either direction, the orifices of both pipes may be fully opened. In the centre of rotation of the valve is an oblong hole, into which is fixed the correspondingly formed end of a long rod *o*, *o*, traversing the whole length of the boiler, and passing steam-tight through a stuffing-box in the back plate of the fire-box. A long lever-handle *p*, is fixed to the outer extremity of this rod, and the engine driver is thereby enabled, with the greatest ease and precision, to regulate the supply of steam to the cylinders. A small pipe *q*, screwed into the upper part of the regulator-valve chest, rises through the smoke-box, and is surmounted by a cup, and provided with a stop-cock, by which oil may be admitted into the interior of the valve-chest, for the lubrication of the working parts.

The two branch steam-pipes J, J, as will be distinctly seen by reference to the transverse section, fig. 15, open a communication for the admission of steam from the regulator-valve chest K, into the valve casing or steam-chest L, L. They are each 3¾ inches internal diameter, and they, as well as the discharge-pipes N, N, are so disposed within the smoke-box as not to obstruct the cleaning or replacing of the tubes.

The Cylinders and Valves.—The slide-valves *r*, *r*, with their expansion slide frames, (which will be fully described hereafter), are placed between the cylinders M, M, in one steam-chest L, formed by the construction of the cylinders when bolted together, as will be seen by inspection of the plans, figs. 6 and 17. By this arrangement access is afforded to both valves by the removal of only one cover, which seems to be an improvement over the other methods now in use.

The steam cylinders M, M, are fourteen inches diameter, with a stroke of twenty-one inches. They are placed at a slight angle in the smoke-box, for the purpose of being accommodated to the position of the cranked axle. The form and dimensions of the pistons P', P', and the arrangement of the packing rings a'', a'', are clearly indicated in the detail sheet, fig. 19. The packing consists of two cast-iron rings a'', a'', turned slightly eccentric, the thick sides in each being set diametrically opposite. At these points they are cut, and wedges b'', b'', fitted accurately into the openings. These wedges are pressed outwards by two springs c'', c'', which are adjustable by set-screws. The whole is rendered compact and secure by the piston cover d'', which is bolted to the body of the piston by four bolts, guarded by the pieces e'', e'', as shown in fig. 19, No. 3.

The steam-ports *s*, *s*, which communicate between each extremity of the cylinders and the slide valves *r*, *r*, are formed in the body of the cylinders, as are also the discharge-ports N, N, to the point where the blast pipes are jointed to them. The discharge or blast pipes N, N, ascend from each cylinder till they reach the bottom of the chimney, where they are formed into one pipe, in the orifice of which is placed a cone or tapered plug *t*, so disposed and connected by means of a system of rods and levers *u*, *u*, as to be capable of being raised or depressed by the engine-driver. By this means the orifice of the blast pipe may be enlarged or contracted at pleasure, thereby causing a greater or less draught to the fire. By this simple contrivance, the engine-driver is enabled to adapt the quantity of steam generated in the boiler to the exact amount required for the supply of the engine, and thereby to prevent the waste of fuel indicated by the steam blowing off at the safety-valve. For the further regulation of the draught when the engine is at rest, it is provided with a damper *v*, at the lower end of the chimney, worked, like the blast regulator, by a system of rods and levers, also marked *v*, *v*, and terminating near the foot plate.

The Framing and Connections of the Engine.—Having described the internal arrangements of the engine, we now proceed to explain the parts by which motion is communicated to the wheels. These are most fully and clearly delineated, in combination, in our sectional elevation, fig. 5, and in the plan, fig. 6. Between the smoke-box and external fire-box, are bolted the four strong malleable-iron beams O, O, O, called the *inside framing*, and which, besides imparting great strength and rigidity to the whole structure, serve the purpose of giving fixed points of resistance for the bearings of the working parts. Of these, the first that claim our attention are the piston-rods P, P. These are made of steel, turned truly cylindrical and smooth, and of the diameter of 2⅛ inches; they are fixed into the piston with a cotter in the manner indicated in the detail fig. 19, and at the opposite extremity they are terminated each by a cross-head Q', also attached to them by a cotter, (see fig. 20). On these cross-heads are bearings for the small ends of the connecting rods Q, Q; and concentric, and of the same piece with these bearings, are projecting arms into which the cast-iron guide-blocks *w*, *w*, fig. 20, are fitted. The guide-blocks are formed with flanges, and are accurately fitted and ground into steel slide-bars, also marked *w*, *w*, so as to work smoothly and steadily between them. These latter are set truly parallel and in the same inclined plane with the centre of the piston-rods, and are firmly bolted to the framing-plates O, O. By this means, the piston-rods are constrained to move in a rectilinear direction, and secured against any deflection, or undue strain arising from the continual change of position of the opposite ends of the connecting-rods, in obedience to the revolution of the cranks to which they are respectively attached.

The feed-pumps S, S, for the supply of water to the boiler, are also set in the line of the piston-rods, and their plungers partake of their motion, being each fixed to a small arm *x*, firmly secured by a cotter to the cross-head Q'. The pumps, the internal arrangement of which is fully shown in the longitudinal section, fig. 24, are formed of cast-iron, and are firmly fastened to the inside framing O, by bolts passing through the projecting flanges f'', f''. The plungers g', are of brass, two inches in diameter, and at each stroke of the engine draw the water from the tender through the feed-pipe *y*, and lower, or suction valve h'', forcing it, at the return stroke, through the upper or delivery valve i'', and along the pipe *z*, into the boiler. The valves are prevented from rising out of their seats by the stops j'', j', fixed into the covers of their respective chests, and so adjusted as to admit of their rising only to the proper height for the due ingress and egress of the water. At the point where the water is discharged into the boiler is placed a valve box a', within which is a valve opening upwards, for the retention of the water within the boiler. A small cock called the *pet-cock* b', is fixed to the outside of the feed-pump, and by means of a long slender rod, the handle is brought within reach of the engine-driver, so that he may be enabled to ascertain at any time whether the pump is working efficiently.

The connecting rods Q, Q, are jointed, as we have before explained, to the cross-heads of the piston-rods. The coupling is effected in the usual way, by means of straps, gibs, and cotters, properly secured against relaxing or falling out. The opposite ends are attached in the same manner to the cranks R, R, upon the axle of the driving-wheels. This cranked axle is made of the best forged iron, the cranks being cut out of the solid mass, and the one formed exactly at right angles to the other. In the earlier stages of the history of the locomotive engine, it was usual to provide bearings for the cranked axle upon each of the frames O, O, but this practice is now discontinued, and thereby the machinery is much simplified, and the friction considerably reduced.

The Eccentrics and Valve Geer.—This engine is provided with four eccentrics c', c', and d', d', the two first for the forward, and the others for the backward geer. The form and dimensions of these are shown upon an enlarged scale in the detail fig. 21, which gives a view of one of the backward eccentrics, but which, with a slight difference to be immediately noticed, presents an accurate type of the whole set. Each eccentric is formed in halves, for the purpose of embracing the axle, and these are joined immovably together by the two round pins k'', k'', screwed into one half, and secured after passing through the other, by cotters. It is fixed firmly to the axle by the two pointed set screws, l'', l''. The forward eccentrics for both cylinders are fixed upon the axle a little *in advance* of a line at right angles to their respective cranks, for the purpose of giving the required *lead*, and the position of the backward eccentrics is adjusted upon the same principle, though of course in a diametrically opposite direction. The eccentric-rods m'', m'', are bolted firmly to the brass strap surrounding the eccentrics; and their opposite extremities, the form of which is shown in fig. 23, are connected together by a double link e', (fig. 22), so formed as to admit of either forward or backward eccentric being thrown into geer with the valve spindle, as may be required. The link which Messrs. Hawthorn employ for coupling the ends of their eccentric rods is of a new and improved construction, being so formed as to diminish as much as possible the friction and wear upon the slide-rod pin and the eccentric-rod ends. The *reversing geer*, or mechanism by which the engine-driver is enabled to propel the engine in either direction, is most fully delineated in fig. 16, and consists of a system of rods and levers f', f', f', commencing with a stud upon the lower extremity of the coupling-link e', and terminating in a long handle, placed in a convenient position near the foot-plate. The motion of the eccentrics is communicated *directly* to the slide-valves by means of the valve-spindles g', g', working through oblong guides at the one extremity, to ensure steadiness, and attached at their opposite ends to the slide-valves r, r, by nuts and jam-nuts, for the purpose of adjustment. We may here take occasion to remark, in anticipation of the subject upon which we are now about to enter, that by Messrs. Hawthorn's arrangement the ordinary slide-valves, when once properly adjusted, never require to be varied, to whatever extent the expansion geer may be employed.

The Patent Auxiliary Expansion Slide-Frame and Geering.—In describing this mechanism, it will be desirable to refer more particularly to the detail drawings on an enlarged scale, figs. 9, 10, 11, 16 and 17, the lettering of which, however, corresponds with that of all the parts represented in the general views. On each of the backward eccentrics is fixed a stud h', (see fig. 21), to which is jointed a rod n'', the other extremity of which is connected with the upper arm of a double lever i', working upon a bearing fixed to one of the framing-beams O, O. The lower arm of this lever is grooved throughout its length to receive a sliding pin, attached by the link j', to a system of rods and levers k', k', k', terminating in a long handle, working on the same centre with the reversing handle f'. The sliding pin is also connected by the rod l', to the hollow spindle m', which works through the stuffing-box of the valve-chest L, and encloses the spindle g', of the ordinary slide-valve.*

The expansion slide-frame n', is worked by the hollow spindle m', being attached to it by means of a slender malleable iron frame, embracing it on all sides, and screwed to the end of the hollow spindle, as clearly indicated in the figs. 9, 10 and 11. It is fitted to, and works upon the same face as, the ordinary slide-valve, but is of such a form as, when the frame is in motion, to overlap alternately the ends of the latter, (the back of the slide-valve being accurately planed and fitted for that purpose), according to the amount of expansion required. This can be varied at pleasure by the mechanism above described; for when the sliding pin which works in the grooved arm i', is brought into the centre of motion of that lever, in which position it is represented in fig. 16, it is obvious that no motion of the slide-frame will ensue, and in this position, when it is not required to work expansively, the geering may be secured so as to obviate all unnecessary wear and tear. If, however, the handle k', be advanced into the position represented in the general elevation, fig. 1, the sliding pin and rod l', which is attached to it, will then be forced downwards, as shown in fig. 5, and the slide-frame will partake of the motion communicated to the lever i', by the backward eccentric, and the amount of this travel will obviously be in proportion to the distance at which the sliding pin is set from the centre of motion. A graduated sector is placed at the foot-plate in view of the engine-driver, as shown in the general elevation, for the purpose of indicating minutely the amount of expansion, or at what part of the stroke the steam is cut off.

It is not our intention, nor would it be in accordance with the plan of these descriptions, to enter into any disquisition upon the application of the expansive principle to the locomotive engine in general, or even to give any lengthened detail of the advantages of the arrangement we have just been describing, in particular. Still, however, we may be allowed to remark that Messrs. Hawthorn's expansion geer appears to us to possess many advantages over most of the other methods hitherto employed. The first and perhaps the most important of these we have before adverted to, namely, the complete independence of the motion of the ordinary slide-valve from that of the expansion-frame, rendering any alteration of the latter, after it has been once properly adjusted, unnecessary, whatever amount of expansion may be employed. The admission and discharge of the steam are thus, in all cases, regular, and take place under the most advantageous circumstances attainable by the ordinary valve. Again, the movement communicated by the backward eccentric to the expansion-slide is so regulated, in relation to that of the ordinary valve, as to produce a very peculiar and advantageous effect in cutting off the steam quickly. Thus, the expansive principle may be employed to any extent, between that due to the cutting off of the steam at $\frac{1}{8}$ of the stroke of the piston, to that which would be produced by the action of the ordinary slide alone, without throttling, or what is technically called *wire-drawing* the steam, a defect so much complained of in most of the other modes of expansion hitherto in use.[*]

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* It may here be remarked, as objections may be urged on the ground of expense, that the hollow spindle is not essentially requisite in this arrangement of valves; the spindle m' may be made solid, similar to the rod of the ordinary slide, and worked through a separate stuffing-box, either above or on one side of it; the mode represented in our Plates is, however, the neatest and most compact arrangement.

The Wheels and Outside Framing.—The driving wheels T, T, are firmly fixed to the cranked axle, the ends of which, produced beyond the bearings, carry the cranks and coupling-rods o', o'. The other extremities of these rods are connected by cranks of exactly the same dimensions with the axle of the fore wheels U, U. By thus connecting the driving and fore wheels, the amount of *traction*, or the surface upon the rails available for the propulsion of the engine, is greatly increased, which renders this species of engine peculiarly suitable for drawing merchandise or other heavy trains, at moderate speeds. The hind or trailing wheels V, V, are situated under the fire-box, and the advantages of this disposition have been already pointed out. The dimensions of all these wheels have also been already given, and the mode of their construction will be clearly understood by reference to the large detail fig. 26, which shows both external and sectional views of one of the driving wheels, but which, as far as regards construction, may be taken as a type of the whole. The nave is of cast-iron, moulded and poured round the arms, which have been previously prepared with a dove-tail at their inner ends, for the purpose of giving additional security. The arms and rim are of the best forged iron, and the latter is accurately turned in the lathe, after being welded together. The tyre, which is also of the best forged scrap iron, is bored internally to a slightly smaller diameter than the rim, and shrunk on. It is then secured to the rim by a few rivets, and the whole turned accurately to the proper form and diameter.

As the whole weight of the engine rests upon the wheels, it may be expected to suffer from jolting in passing over the irregularities of the rails. To obviate this as far as possible, the springs p', p', p', and q', q', q', are interposed, the former upon bearings in the outermost of the internal framings O, O, and the latter under the axle-boxes r', r', r', of the main external bearings. The springs marked q', q', and the mode in which they are attached to the axle-boxes and to the framing, are clearly represented in the detail figs. 27 and 28. They are composed of thin layers of steel, gradually diminishing in length from the centre to the extremities, and bound together by the connecting hoop o'', secured in its place by a small round pin passing through it and the steel plates. The connecting hoop is formed with a tail projecting upwards into the lower portion of the axle-box, where it is fixed by a round pin p'', passing through it. The axle-box r', which is of cast-iron, fitted with bearings composed of a metallic alloy favourable for the reduction of friction, (Fenton & Bott's patent), slides up and down as the springs bend with the weight of the engine, between the cast-iron axle-guides q'', q'', which are accurately planed and fitted to receive it, and bolted firmly to the plates of the external framing. The axle-boxes are formed with a sort of reservoir for oil or tallow, which is constantly supplied to the rubbing surfaces by two small tubes and syphon-wicks. It may here be remarked, that the other rubbing parts of the engine are lubricated in the same manner. The mechanism by which the springs are attached to the external framing is shown in fig. 28. These parts are called the *spring-links*, and consist of a species of small cross-head r'', fitted with round pins for passing through the plates of the external framing, and with screwed studs attached by similar round pins to the ends of the springs q', q'. The nut s'' works into these screws, and by means of it the weight which it may be thought expedient to throw upon each spring may be accurately adjusted.

The external framing consists of two strong parallel beams W, W, extending somewhat beyond the engine at both ends, and connected in front by the wooden cross-beam or *buffer-bar* Z, and behind by a similar beam, on which rests the foot-plate Y. These beams are firmly bound together at the corners by angular plates of iron bolted through each, and the weight of the boiler is supported upon them by the strong malleable iron brackets or stays X, X, X, riveted to the boiler and bolted through the beams W, W. These latter are formed each of two parallel plates of iron, cut out into the form shown in the general elevation, with *horns* projecting downwards for the bearings of the wheels. Between each pair of plates a beam of well-seasoned oak is interposed, and the whole firmly bolted together.

To deaden the shocks to which the engine is exposed, it is provided with *buffers* s', s', fixed to and projecting in front of the buffer-bar Z. These buffers are a species of elastic cushions, formed of horse hair, surrounded by strong leather, and further strengthened by slender malleable iron hoops. To secure the engine against the effects of the wheels coming in contact with stones or other obstacles which may happen to be lying on the rails, it is furnished with strong malleable iron *safe-guards* t', t', descending from the external framing to within a short distance of each rail, and so formed at the points as to turn aside any object with which they may come into collision.

Any water which may happen to accumulate in the cylinders, whether from the priming of the boiler or the condensation of the steam, and which, unless removed from time to time, would be very detrimental to the working of the engine, is let off by means of the pipe and stop-cock u', communicating with the discharge passage of each cylinder.

Upon the front of the smoke-box and towards the top, is fixed a small bracket for supporting the *signal lamp* v', by which notice is given at night of the approach of the engine and train.

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[*] These theoretical advantages have been fully corroborated by the results of experience. Messrs. Hawthorn have successfully applied their expansion geering to the locomotives which they have supplied to ten different railways in this country and on the Continent, and the saving of fuel effected by the use of it, in many cases, amounts to 30 per cent.

As a precaution against accident to the engine-driver and his assistant, hand-rails w', w', are erected on each side of the foot plate Y, and these are continued along the whole length of the boiler, so that they may be enabled with comparative safety to walk round the engine, even when it is in rapid motion. The rods forming this latter part of the hand-rail are made hollow, and thus afford a neat and compact guide and protection for the slender rods by which the blast regulator on one side, and the damper on the other, are worked.

When the engine is at rest, the steam which would otherwise escape at the safety-valve and be thrown to waste, is made available for the heating of the water in the tender. This is accomplished by means of the bent pipe x', by which a communication is made between the steam within the fire-box and the feed-pipe *y*, and thereby a considerable saving of fuel is found to be effected.

The connection of the engine and tender is made by means of the strong double link or *drag-bar* y', one end of which is secured by a strong pin to a bracket fixed under the foot-plate of the engine, while the other is in a similar manner jointed to the drag-springs of the tender.

To assist the engine-driver in rising into his place on the foot-plate, the footsteps z', z', depend on each side of it to within an easy distance from the ground.

Having thus minutely described the parts of which this engine is composed, and explained their several uses as we went along, we consider it unnecessary to occupy more space with an account of its mode of action. This is in every respect identical with that of the ordinary high-pressure engine, and to those who have followed us in our previous descriptions, will be perfectly intelligible.

Description of the Tender.

The tender is an invariable concomitant of the locomotive engine, and as in it, as well as in the engine, there is considerable room for the display of tasteful design and judicious arrangement, we have thought that we should render our engravings more interesting and more acceptable, by giving representations of both. The water tank A, A, forms the principal part of the tender, and consists of a rectangular sheet-iron cistern, capable of containing 1200 gallons of water for the supply of the boiler. It is made with a long recess B, for the reception of coke, the species of fuel which is mostly in use for locomotive engines. The floor *a*, of this recess, is made with a slope downward from the front of the tender, by which arrangement the fuel is prevented from being thrown out by any jolting or shaking to which it may be subjected.

Towards the back of the tank it is surmounted by a pipe or opening C, by which water is introduced from the water-crane or other contrivance for that purpose. A wooden cover is fitted over this opening when not in use. At the same point are fixed the wooden tool-boxes D, D, for containing spanners and other implements which may be required for the engine. At the front of the tank, and on each side, are situated the cocks *b*, *b*, for regulating the supply of water to the suction-pipes *c*, *c*, communicating between the feed-pumps and the tender. These are connected to the feed-pipes *y*, *y*, by means of leather hose, screwed on to each by the union joints *d*, *d*, thus admitting of a considerable amount of vibration or change of position of the pipes, without breaking the connection. The tank is secured to a strong wooden frame E, E, forming the body of the tender, and strengthened by numerous cross beams. Beyond this wooden framing, and on each side of it, the external iron framing-plates F, F, are fixed by bolts passing through short cross beams of timber *e*, *e*, *e*, abutting against both. The external framing-plates are made of a form symmetrical with those of the engine, as seen in the general elevations, figs. 1 and 3, and their purpose is to afford bearings for the reception of the axle-boxes *f*, *f*, *f*, which slide up and down in them in obedience to the action of the springs *g*, *g*, *g*.

The tender is supported upon six wheels G, G, G, of the same diameter as the trailing or hind wheels of the engine, and constructed in the manner already described in treating of the latter. The brake apparatus, which is shown on an enlarged scale in fig. 29, consists of a train of mechanism by which a great amount of friction can be simultaneously produced upon the peripheries of the tender wheels, for the purpose of reducing the momentum of the engine and train, when it is required to arrest the motion of the train. The hand-wheel *h*, is fixed to the upper extremity of a vertical spindle H, working in a strong bearing attached to the tank. The lower portion of the spindle is formed into a screw, and works through the wrought-iron nut I, on which is forged a double link, jointed at its lower end to the brake-lever *i*. The latter has its centre of motion in the short shaft J, which works in strong bearings attached to the wooden frame, and carries the double toothed sector *j*. Two longitudinal iron rods *k*, *k*, extend the whole length of the tender, and a small portion of each towards the front extremity is formed into a rack, so adjusted as to work into the teeth of the sector *j*. The rods *k*, *k*, are supported and guided in their motion by small rollers working in the wrought-iron guides *l*, *l*, *l*, and upon them are bolted the wooden brake-blocks *m*, *m*, *m*, by the contact of which with the exterior surface of the wheels the friction is produced. By this arrangement it is obvious that by screwing the vertical spindle H into the nut I, the latter will be drawn upwards, and carrying with it the lever *i*, the toothed sector *j*, will be made to revolve upon its axis J, and consequently the rods *k*, *k*, will be drawn each in the opposite direction to the other. Each wheel will therefore be forcibly compressed between the brake-blocks *m*, *m*, and the engine and train be proportionally retarded.

At the point where the engine is connected to the tender, the latter is provided with a system of springs, to deaden the effects of shocks from either direction. This consists of two springs set back to back, and connected together by a socket *n*, which receives the end of the drag-bar. The fore spring *p*, comes into action when any force is applied tending to *separate* the engine from the tender, as in starting a train, and the hinder spring *o*, when the force is applied in the opposite direction. Both springs are supported upon pieces of thin iron, bolted between the beams of the wooden frame, and the extremities of the spring *o*, bear upon the two guide-pins *q*, *q*.

For further security, in case of the ordinary connections failing, the safety-chains *r*, *r*, are attached between the engine and tender.

For the accommodation of the engine-man and stoker, the tender is furnished with foot-steps *s*, *s*, placed at an easy distance above the steps of the engine. By these arrangements, and with the assistance of the handles *t*, *t*, the foot-plate is rendered easily accessible.

At the front of the tender, a piece of boiler-plate *u*, is fixed by hinges, for the purpose of forming a floor where the engine and tender are connected. At the other extremity of the tender, the buffers *v*, *v*, similar in construction and in situation to those formerly described, are fixed to the cross beam of the wooden framing, for the purpose of deadening the shocks produced by the occasional irregularities of motion between the engine and the train. The drag-chain *w*, which is firmly secured to the same beam, forms the connecting link between the tender and the train.

Literal References to the Engine.

A, the external fire-box.
B, the internal fire-box.
C, C, stays for strengthening the roof of the internal fire-box.
a, *a*, *a*, stays between the external and internal fire-boxes.
b, the fire-door.
c, *c*, the fire-bars.
d, the movable portion of the fire-bars.
D, the ash-box.
E, the cylindrical part of the boiler.
e, *e*, *e*, the tubes.
f, *f*, longitudinal stays from the back of the fire-box to the front of the boiler,
F, the smoke-box.
g, *g*, the smoke-box doors.
G, the chimney.
H, a brass funnel for enclosing the safety valves.
h, the spring safety valve.
i, the lever safety valve and spring balance.
j, the water gauge and gauge cocks.
k, the steam whistle.
l, the blow-off cock.
I, the steam receiver.
m, the inverted cone for preventing priming.
J, J, the steam pipes.
K, the regulator valve chest.
n, the regulator valve.
o, a rod connecting the regulator valve with
p, the handle for working it.
q, the oil cup and pipe for lubricating the regulator valve.
L, the steam chest of the cylinders.
r, *r*, the slide valves.
M, M, the steam cylinders.
s, *s*, the steam ports.
N, N, the discharge ports and blast pipes.
t, the blast regulator.
u, *u*, handle, rods, and levers for working the blast regulator.
v, *v*, the damper, with the handle, rods, and levers for working it.
O, O, the inside framing of the engine.
P', the steam piston.
a'', a'', the packing rings of the piston.
b'', b'', wedges for tightening the packing.
c'', c'', springs bearing on the back of the wedges b'', b''.
d'', the piston cover.
e'', e'', guards for the bolts of the piston cover.
P, P, the piston rods.
Q', Q', cross-heads for the piston rods.
w, *w*, the cross-head slides.
x, *x*, projecting arms for working the feed pumps.
Q, Q, the connecting rods.
R, the cranked axle.
S, S, the feed pumps.
f'', f'', flanges for bolting the feed pumps to the inside framing.
g'', g'', the plungers of the feed pumps.
h'', lower or suction valve of the feed pump.
i'', upper or delivery valve.
j'', j'', stops for regulating the lift of the valves.
y, *y*, the feed pipes from the tender to the feed pumps.
z, *z*, branch pipes from the feed pumps to the boiler.
a', a', valve boxes at the boiler.
b', b', the pet cocks and their handles.
c', c', the forward eccentrics.
d', d' the backward eccentrics.
k'', k'', bolts for connecting the halves of each eccentric.
l'', l'', steel pinching screws for fixing the eccentrics to the axle.
m'', m'', the eccentric rods.
e', e', coupling links for the ends of the eccentric rods.
f', f', f', levers, shafts and rods for working the reversing geer.
g', g', the main steam-valve spindles.
h', h', studs on the backward eccentrics, for working the expansion slide frames.
n'', n'', connecting rods between the studs h', h', and
i', i', the grooved arms for the variable expansion.
j', j', links between the grooved arms and the levers k', k'.
k', k', k', levers, shafts and rods for regulating the expansion geer.
l', l', connecting rods between the grooved arms i', i', and
m', m', the hollow spindles attached to
n', n', the expansion slide frames.
T, T, the driving wheels.
o', o', the outside cranks and coupling rods.
U, U, the fore wheels coupled to the driving wheels.
V, V, the hind wheels under the fire box.
p', p', springs for the inside bearings of the cranked axle.
q', q', springs for the outside bearings of all the axles.
o'', o'', connecting hoop for the outside springs of the cranked axle.
r', r', the axle boxes.
p'', p'', pins for attaching the springs to the axle boxes.
q'', q'', cast-iron guides for the axle boxes.
r'', r'', the spring links.
s'', s'', the nuts for adjusting the weight upon the springs.
W, W, the external frame of the engine.
X, X, stays from the external frame to the boiler.
Y, the foot plate.
Z, the buffer beam.
s', s', the buffers.
t', t', the safe-guards.
u', u', a cock and pipe for letting off water from the cylinders.
v', the signal lamp.
w', w', the hand railing.
x', a pipe from the boiler for heating the water in the tender.
y', the drag bar.
z', z', the foot steps.

Literal References to the Tender.

A, the water tank.
B, the recess for containing the coke.
a, *a*, the floor of the coke box.
C, the opening into the tank.
D, D, the tool boxes.
b, *b*, the cocks for regulating the supply of water to the feed pumps.
c, *c*, water or suction pipes to the engine.
d, *d*, union joints for connecting the feed pipes.
E, E, wooden frame of the tender.
e, *e*, stays between the wooden and iron frames.
F, F, the iron frame for receiving the axle boxes.
f, *f*, *f*, the axle boxes.
g, *g*, *g*, the springs.
G, G, G, the wheels.
H, the vertical spindle and screw for working the brake.
h, the hand wheel for the brake screw.
I, the nut and link for connecting the screw with
i, the brake lever.
J, the short shaft carrying the brake lever, and
j, the double-toothed sector, working into
k, *k*, the longitudinal rods carrying the brake-blocks.
l, *l*, supports fitted with rollers for guiding the rods *k*, *k*.
m, *m*, the wooden brake-blocks.
n, socket for connecting the drag springs to the drag bars.
o, *p*, the springs for buffing and drawing.
q, *q*, bearings for the spring *o*.
r, *r*, the safety chains.
s, *s*, the foot steps.
t, *t*, handles to assist in rising to the foot-plate.
u, a hinged plate between the engine and tender.
v, *v*, buffers for the tender.
w, drag chain of the tender.

PLATES LXXIV.—LXXVII.*

DIRECT ACTION ENGINES OF H. M. STEAM FRIGATE DRAGON, OF 560 HORSE POWER COLLECTIVELY. BY WILLIAM FAIRBAIRN AND SONS, LONDON.

We have already, in former parts of this series, (Plates LVI.—LXIII.) given very full illustrations and descriptions of that class of marine engines denominated the *side-lever* engine. In the Plates now under notice, we present to our readers an excellent example of that other class called, in contradistinction to the former, *direct-action* engines. The distinctive feature of this species of engine consists in the line of shafts being placed immediately over the centre of the cylinders, and in the motion of the piston being transmitted *directly* to the crank by the connecting rod, without the intervention of the side-lever. By this arrangement the space which the engines occupy in the vessel is very much diminished, while at the same time their weight is reduced by at least one-third as compared with those on the side-lever principle. But on the other hand, these advantages are not gained without incurring a certain loss of effect consequent upon the disposition of the working parts of the direct-action engine, and however the amount of this loss may vary in different engines of this class, there will still remain some irremediable defects common to them all. Of these the most obvious are the apparently unavoidable shortness of the stroke, and of the connecting rod, with the consequent loss of power incurred by the application of the force of the engine to the crank at a considerable angle, and the increase of friction generated in preserving the parallelism of the piston rod; also a deficiency in the balancing of the moving parts, producing irregularity of motion and increased wear and tear. Notwithstanding these defects, however, their superiority over the side-lever engines, in regard to stowage and weight, is so great as to have caused their very general adoption in steam-vessels of every description, and especially in the navy, for which service they have been found to be peculiarly adapted. Besides the Dragon, Messrs. Fairbairn have already fitted the two Government steam-ships, Vulture and Cormorant, with engines upon the same plan, which have all proved most efficient, and a fourth pair are now in progress for the steam-frigate Odin, of 560 horse-power. In all these engines the defects above noted are, in a great measure, obviated by the increased strength and dimensions of the parts; and the precautions taken by Messrs. Fairbairn in the enlargement of the bearings give assurance of great security, and a proportionate diminution of friction in all the parts.

Enumeration of the Plates.

Plates LXXIV. LXXV. represent a side elevation of one of the engines, as seen from the centre line of the vessel. The position of the piston has been taken at the half descending stroke, as shown by the *lead* of the crank, by which also it will be observed that the steam-pipes are introduced at the after end of the engines.

Plates LXXVI. LXXVII. represent sectional plans of the two engines, as they are disposed in the vessel. The one engine is shown in a horizontal section taken through the cylinder, air-pump, and condenser, with the plan of the entablature, cranks, &c., shown in dotted lines. The other engine is represented as sectioned immediately under the entablature, in order to show most clearly the general disposition of the working parts.

Plates LXXVI.* LXXVII.* are occupied with details of some of the principal parts of the engine detached.

Fig. 1 represents the paddle shafts from the centre line of the vessel to the outer bearing. In this figure are introduced a section of one of the cranks, and of the cam for working the expansion valve.

Fig. 2 is a vertical section, (taken through the centre of the valve-face), of the cylinder and piston, with the piston rod and cross-head.

Fig. 3 shows a plan and sections of the elastic packing ring at the middle and ends.

Fig. 4 is a sectional view of the steam-pipe, showing the expansion and throttle valves. Fig. 5 represents a vertical section of the expansion valve detached.

Fig. 6 is a transverse section of the air-pump and feed-pump with their appendages. In this figure are also introduced external views of the air-pump rod and cross-head, and of the plunger-rod of the feed-pump.

Figs. 7 and 8 show the joint or link for conveying motion to the air-pump cross-head.

Figs. 9 and 10, the connecting rod.

Figs. 11 and 12, the main parallel motion beams, which also work the air-pump.

Fig. 13, the radius rod for the parallel motion.

General Description.

The detailed description already given in our explanation of Plates LVI.—LXIII. is to a great extent applicable to marine engines of every kind, and need not here be recapitulated. We propose, therefore, under the above head, to give only such particulars regarding the engines now before us, as could not be introduced into our *literal references*, which, for the sake of pointing out the peculiarities of the Dragon's engines, shall be made fuller than usual. Under this last head also we shall note the principal dimensions of the parts to which reference is made, and it is hoped that these, together with the dimensions given in the detail Plate of the present series, will render the whole more useful and acceptable to the practical Mechanic.

In order to save weight, the Dragon's framing is made entirely of wrought-iron, with the exception of the entablatures. It will be seen from Plates LXXVI., LXXVII., that the engines, when in conjunction, are arranged in the form of a compact square, being thus condensed within the smallest practicable limits. The space they occupy collectively is 13 feet 6 inches fore and aft, and 20 feet 6 inches athwartships. The nominal power of each engine is 280 horse, computed by the Government formula, which for engines of this size is as follows, viz.:—

$$\frac{\text{Area of piston} \times 7\text{ lb.} \times 220\text{ feet.}}{33,000} = \text{horse power.}$$

The parallelism of the piston rod is maintained by the strong parallel motion beams K, K and L, L, the latter being extended to work the air-pump. The cylinder valves employed are on the *long D* construction, in which the steam is admitted round the hollow valve and exhausts through it.

The following are some of the leading dimensions which are not given in figures in the Plates:—

Dimensions and Distances.

	Ft.	Ft.	In.
Length of the engine room,................................	52	60	0
— coal hold,....................................	8		
Breadth in the clear,..		34	4
Depth, — —		20	0
Load draught of water of vessel,.............................		16	0
Centre of shaft above load line,...............................		8	6
Diameter of paddle wheels to outer extremities of boards,		27	0
Breadth of boards,..		10	0
Depth of do.		2	3
Strokes per minute, (estimated number).....................		19	
From under side of base-plate to centre of shaft,..........		20	4
— centre to centre of engines,..............................		12	4
— centre of cylinder to centre of air pump,.............		6	5⅝
— centre to centre of columns, fore and aft,...........		8	4
— — — — athwartships,...........		5	7½

Diameters of Pipes.

	In.	
Steam pipes for each cylinder, (of copper),..................	19	bore.
Injection pipes, — — —	6	—
Feed pipes, — — —	5½	—
Bilge pipes, — — —	5½	—
Waste water or discharge pipes, —	21½	—
Air pipes on do. do. —	5¼	—
Hand pump, connected to the engine, has suction and force pipes,...	4¼	—
Supplementary engine, steam pipe,............................	3¼	—
Waste steam pipe for the boilers,...............................	18$\frac{5}{4}$	—
Blow-off pipes, do. do.	4½	—

Boilers.—The Dragon's boilers are formed on the tubular construction; they are three in number, and measure over all 19 feet 4 inches fore and aft, and 24 feet 8 inches athwartships. They are fired by seven furnaces at each end, fore and aft, the fire-bar surface being 78 square inches per horse-power, and the total heating surface in tubes and fire-boxes, 14·75 square feet per horse. They contain 1200 iron tubes, 3 inches diameter, and are provided with brine-pumps and refrigerators, small engine for feeding, separate steam-pipes and stop-valves, safety-valves, vacuum-valves, and all the usual appendages.

Weights of Engines, Boilers, &c.

Weight of the engines and paddle shafts,....................	174	tons.
— — paddle wheels,......................................	32	—
— — coal boxes,..	18	—
— — duplicates and spare geer,....................	26	—
— — boilers,..	65	—
— — water in boilers,...................................	55	—
Total weight of machinery,............................	370	tons.

Coals stowed in boxes, 445 tons, being 12½ days' consumption.

Literal References.

A, the sole plate, of cast-iron 2 inches thick, laid on a solid bed of African oak 16 inches thick, and cast so as to form the bottom of the cylinder. It is made of larger dimensions than are absolutely required for the attachment of the fixed parts of the engines, being extended to some distance on each side in order to cover a larger space, and to increase the strength of the kelsons and lower timbers of the ship. The sole plate does not, however, on that account, occupy space which could be available for other purposes, being so constructed as to form the flooring all round the engines, and at the same time to distribute their weight and action over a more extended surface.

B, B, the four main columns of the framing, formed of wrought-iron, 7 inches diameter. They are turned and keyed into sockets cast on the sole plate and the entablature, but pass loosely through the bored sockets on the cylinder, in order to allow for the alternate expansion and contraction of the latter by variations of temperature.

C, the entablature, of cast-iron, the upper part of which forms the crank-shaft plummer-block. It is cast in the form of a hollow box, and is securely keyed to the heads of the columns, which fit into bored sockets, 30 inches long. The entablature is tightly wedged between teak beams, 30 inches by 20, which receive the thrust of the engine, and from their great inertia intercept the vibrations produced by the action of the paddle wheels.

a, the crank-shaft plummer-block cover.

D, D, the great cross stays of the framing, with bosses at *b*, *b*, for the fixed centre on which the parallel motion beams vibrate, and one at *c*, for the transverse stay between the two engines. These stays are formed of forged iron, 2¾ inches thick, the sockets being truly bored to the diameter of the columns, and secured to them by keys.

E, the steam cylinder, 1¾ inch thick, cast open at each end, the sole plate forming the cylinder bottom. Its internal diameter is 88 inches, and the stroke 5 feet 9 inches. Besides being strengthened by the flanges at top and bottom, it is encircled by three belts, each three inches deep, in order to give rigidity to the cylinder when laid horizontal during the process of boring. It is now usual to bore cylinders of large diameter in a vertical position, to avoid all chance of ellipticity arising from this cause. The cylinder is fitted with escape valves for priming or condensing water, both above and below the piston, the valve being loaded to somewhat above the highest tension of the steam in the cylinder.

d, *d*, the steam ports, 3 feet 8 inches long, by 7⅞ inches wide.

e, a separate casting for the cylinder face, planed and scraped to a true surface.

F, the valve casing, of cast-iron, ⅞-inch thick.

f, the steam valve, of the construction usually called the *long D valve*, formed of cast-iron, ⅞-inch thick, strengthened by two ribs ⅝-inch thick, and faced with brass ½-inch thick, planed and scraped.

G, the main steam pipe, of copper, 19 inches bore.

g, the expansion valve seat, of gun metal.

h, the expansion valve, of the same material. This valve is formed on the principle of the Cornish or equilibrium valve, and is so named from its action being uninfluenced by a pressure of steam, as it is moved with equal facility in any position. Its value consists in its giving the means of effecting a large opening, with a small amount of motion, and at the same time requiring little force to work it, while large valves of the common construction are heavy, or are so much pressed in one direction by the steam as to require a considerable expenditure of power in working them. The valve seat *g*, consists of an upright ring or collar fixed in a chest or chamber prepared to receive it, and which is merely an expansion of the steam pipe G. This ring rises to a considerable height in the chamber, and is perforated on all sides by numerous large slits or openings, but is closed at the top. The valve *h*, is likewise in the form of a ring, close all round the sides, but open at the top, where it is attached to the rod or spindle by which it is raised or depressed. Both rings are accurately turned and ground to fit into each other, and are provided with bearings which, when the valve is depressed, prevent the passage of steam through it. When, however, the valve is raised, the steam entering through the top of the crown or cover *h*, passes through the apertures of the ring *g*, and thus a free communication is effected between the upper and lower sides of the valve, with little greater expenditure of power than what is due to the friction of the parts.

i, the throttle valve, of gun metal, worked by one of the three handles at *v′*, through a series of rods and levers.

H, the steam piston, 8⅜ inches deep at the circumference, and 11¾ inches at the centre, formed of cast-iron 1⅓ inch thick, and strengthened by radiating feathers.

j, *j*, the external packing rings, of cast-iron, 2½ inches deep, on Goodfellow's patent construction. These rings are turned concentric, and each, being cut across diagonally, to prevent grooving the cylinder, is free to accommodate itself to the interior of the cylinder as it wears by the friction of the piston, but possesses no elasticity in itself.

k, and *l*, represent the thick and thin ends of the inner eccentric V-shaped ring, of cast-iron, which by its elasticity presses the two external rings against the cylinder, thus forming a steam-tight joint. The requisite elasticity is given to this ring by forming in it a series of cuts or grooves increasing regularly in depth from *k*, where the ring is entire, to *l*, where the grooves are deepest; and also by causing the projections at the back of the rings shown at *k*, to vanish at *l*. The ring is 4¾ inches deep, its thickness being 5⅝ inches at the thickest part, and ⅝-inch at the thinnest. The grooves (86 in number) are, at the point *l*, 2⅜ inches deep, and 7/16-inch wide, and they are pitched at 2½ inches centre to centre.

m, the junk ring, of cast-iron, 1½ inch thick.

n, a brass nut for fixing the piston rod.

I, the steam cylinder cover, cast hollow, with radiating feathers, and with a thin polished plate of cast-iron for the upper part.

o, the stuffing box gland, made so as to allow the strap of the connecting rod to clear it on each side. It is formed of cast-iron, with a brass lining, and packed with hemp.

J, the piston rod. It is fastened into the piston by a cone and screw *n*, and further secured by a cotter.

p, the piston cross-head, of forged iron, secured to the rod by a long cone and cotter. The bearings for the connecting rod and the first parallel motion beams are forged in one piece with it.

K, K, the first parallel motion beams, vibrating on the hanging centre *t*, are formed of wrought iron, 2⅜ inches tapered to 2⅛ inches thick. The distances between the centres are 3 feet 9⅝ inches, and 2 feet 9 13/16 inches. The depth of the beam at the centre is 15¾ inches, and at the ends 8 inches.

q, the vibrating shaft for the parallel motion. It is made of wrought iron, 6 inches diameter at the bosses, and 5 inches diameter at the necks, which are 5⅜ inches long.

r, *r*, radius rods for the parallel motion.

s, a wrought iron stay-rod for strengthening the fixed bearing of the rod *r*.

t, the hanging centre for the parallel motion beams. The diameter of the bosses is 8¼ inches, and that of the necks 6½ inches by 6½ inches long.

u, the fixed centre for the second parallel motion beams.

L, L, the second parallel motion beams of wrought-iron, extended to work the air pump. Their thickness is 2⅜ inches, tapered to 2⅛ inches, the thickness at the centre boss being 11 inches, and at each end 5 inches. Diameter of bearing in centre of beam, 8 inches, and at each end 6½ inches.

v, a cross stay of wrought iron, between the ends of the beams L, L, and from which the air pump links are hung. The necks are 5⅝ inches diameter.

The parallel motion formed by the combination of the parts above detailed is adopted not only to guide and direct the perpendicular ascent and descent of the piston, but also to effect the working of the air pump as well as of the feed and bilge pumps.

M, the connecting rod, formed with a fork at the lower extremity, which is attached by straps, gibs and cotters to the great bearings on each side of the piston cross-head *p*.

N, the crank, of wrought-iron.

w, the crank pin. That portion of it which passes through the crank on the intermediate shaft P, is firmly fixed to it by being driven into a slightly tapered eye, and is further secured by a cotter; but at the opposite extremity it passes loosely through the crank on the paddle shaft, to allow for the deviation from the true vertical and horizontal lines of the shafts, occasioned by the unavoidable tendency of the outer bearing of the paddle shaft to droop and to wear forwards.

O, the paddle shaft of wrought-iron.

x, *x*, *x*, bosses on the paddle shaft for fixing the paddle centres.

P, the intermediate shaft which connects the two engines. The diameter at the centre is 18½ inches, and the bearings are 17 inches diameter, by 18 inches long.

y, the position of the eccentric. In this as in most other marine engines, the eccentric is not fixed firmly to the shaft, but revolves between the bearings shown at *y*, to admit of its working the valves so as to give the engine motion either backwards or forwards. By placing the eccentric loose upon the shaft, only with a projecting catch on one side which is carried round by a corresponding projection on the shaft, it is left free, except when this check comes into contact with the other, at either end of the stroke. To effect this it is necessary to open the valves by hand during at least one half stroke.

Q, the eccentric, of cast-iron, revolving within a ring of gun-metal. The throw of the eccentric is 20 inches; the diameter of the bearing surface is 3 feet 5⅝ inches; the thickness of the gun-metal ring is 1¾ inch, and the breadth 4 inches.

z, the back balance of the eccentric.

R, the eccentric rod, forged in a rectangular form. Its depth above the gab is 6¾ inches to 5¾ inches, and below this, 4 inches to 2½ inches; the breadth of face is 3¾ inches.

a′, a projection on the eccentric rod, into which a corresponding projection on the eccentric ring is fixed with a cotter, for the purpose of giving greater steadiness to the motion of the eccentric rod.

b′, the stud on the valve lever for receiving the eccentric gab. Its diameter is 4 inches, by 3⅞ length of bearing.

S, the valve or traverse lever, of wrought iron.

c′, a wrought-iron shaft, keyed into bosses forged on the inner cross stays D, D, of each engine, and consequently forming a connection between the two engines. Its principal use is to afford a centre of motion for the valve levers S, S, which vibrate loosely between collars forged upon the shaft *c′*, but from its position and strength, it also serves the very important purpose above adverted to.

d′, the valve links, of wrought iron, with solid eyes bushed with brass. The bearings are 2⅞ inches diameter, and 2⅝ inches long.

e′, the parallel motion rods for the valve links, 2 inches diameter in the centre.

f′, the valve spindle, 3 inches diameter, with a cross-head adjustable to the proper height by a screw and nuts above and below.

g′, parallel motion rods for the back balance of the valves.

T, the back balance to the steam valve, having a rack cast upon it, into which works a small pinion *k′*, on the axle of the starting wheel.

h′, a guide rod and bearing for the back balance, the rod descending into an open pipe sunk in the condenser.

i, a lever with brass roller, for disconnecting the eccentric rod, and throwing the starting wheel into geer.

j′, the starting wheel, 6 feet diameter.

k′, a pinion, 8 inches pitch diameter, on the axis of the starting wheel.

l′, a column with forged bearings for the starting geer.

m′, diagonal stay rods for strengthening the column *l′*.

n′, a catch for retaining the lever *i′* in the position into which it is set.

The starting geer, of which the above are the detailed parts, is a very beautiful and effective piece of mechanism, and, with various modifications, is now very generally superseding the old system of single and separate levers. Its action will be understood by reference to the general elevation, in which the eccentric rod is represented in geer with the valve lever and the starting geer, of course, disconnected. By depressing the end of the lever *i′*, the eccentric rod is disengaged from its connection with the pin *b′*, and at the same time the pinion *k′* is thrown into geer with the rack upon the back balance T; but the mechanism is so arranged that this latter connection is not effected until the eccentric rod has cleared the pin, so that the starting wheel can never be driven round by the motion of the valve levers.

o', the air-pump links. Thickness of strap, 1⅜ inch; the column 2⅜ inches tapered to 1⅝ inch diameter.

p', the air-pump cross-head. The diameter over the boss is 11⅛ ins.; diameter of hollow cone, 5$\frac{7}{16}$ inches tapered to 5⅛ inches; thickness of blade, 3 inches.

q', the air-pump rod, formed of wrought-iron cased with gun-metal ½-inch thick when finished, cast round the rod and the cone, to prevent galvanic action.

r', the air-pump bucket, of gun-metal, packed with hemp.

s', s', the bucket valves, of gun-metal, of the description commonly called *butterfly valves*. They work on a curved hinge encircling the pump rod.

t', t', discharge valves, also of gun-metal, and of the same construction as the bucket valves.

U, the condenser, of cast-iron, 1¼ inch thick. It is cast in one piece for both engines, with a partition through the middle to keep the condenser of each distinct from the other. There is a passageway through the lower part of the condenser to the valve jackets, in order to give access to the valve ports for the purpose of introducing the packing.

V, the air-pump barrel, of solid gun metal, ¾-inch thick when finished, 48 inches diameter, and 34½ inches stroke.

W, the feed-pump, with brass plunger, 9½ inches external diameter, and 34½ inches stroke.

X, the bilge-pump, of the same dimensions as the feed-pump.

u', u', the feed and bilge-pump rods, 3 inches diameter, which serve also as guides to the air-pump cross-head. As it is the practice in engines for sea-going vessels to make each feed-pump sufficiently large to supply all the boilers in case of derangement of one pump, and as the bilge-pumps are not required to be constantly at work, it becomes necessary to provide a ready mode of connecting and disconnecting the pumps whilst the engines are in motion. This is effected by means of a brass bush fitting loosely in the eye of the air-pump cross-head end, through which the rod passes freely when the pump is disconnected; but when it is required to connect the pump, the pressure of a screw is applied to the brass bush, until the friction exerted is sufficient to hold or *clip* the rod, and consequently to give it the motion of the air-pump cross-head. It is apparent, that whether the feed-pump be at rest or in motion, it acts equally as a guide to the air-pump cross-head, the only difference being, that in the one case the *spindle* forms the guide, and in the other the *plunger*.

Y, the discharge pipe, of cast-iron, forming the top of the air-pump. From the air-pump to the sluice-valve through the ship's side, it becomes a circular copper pipe 21½ inches bore.

Z, the supplementary engine for feeding the boilers previous to starting the large engines.

v', v', v', a combination of levers and rods for working

w', the injection valve.

x', the blow through or shifting valve, and x', the throttle valve. The connections for each valve are distinctly shown in the elevation.

y', a graduated cam on the intermediate shaft for working the expansion geer.

z', z', z', a brass roller, levers and rods for conveying the motion from the cam to the expansion valve. The roller is adjusted to each cam by a screw, according to the amount of expansion required.

PLATES LXXVIII.—LXXX.

HORIZONTAL SUGAR MILL, BY M. NILLUS OF HAVRE.

These Plates furnish a good example of the form of machine used for crushing sugar canes, as constructed by an eminent French engineer, who has devoted his attention, in an especial manner, to the improvement of the apparatus used in the colonies for the manufacture of sugar. It differs but slightly from the form usually adopted by our English makers; and the only reason for our having chosen it in preference to a sugar mill of British manufacture, is, that in the latter we conceive that a more elegant and tasteful form of framing, combined with greater economy of material, as exemplified by the French maker, might be adopted. Many improvements have been recently proposed, and successfully introduced into this branch of mechanical practice, such as those embodied in the patents of Mr. James Robinson of London, the object of which is principally the multiplication of the pressure to which the canes are subjected, by increasing the number of the rollers. But although we have no doubt of the perfect efficiency of these machines, we have preferred giving engravings of the more simple and compact form which is still mostly in use.

The conformation of the sugar cane does not render it necessary that, for the extraction of its juice, the cells which contain it should be previously broken, as is the case with sugar obtained from other sources; simple pressure, properly applied, is all that is required for its expulsion. For this purpose the canes are squeezed by being passed successively between rollers disposed somewhat like those of a rolling mill. In the older form of machines employed in our colonies for the extraction of the juice, the rollers are placed vertically, and it is only within the last few years that this arrangement has been superseded by the sugar mill with horizontal cylinders, which is not only cheaper in construction and more easily fixed, but by its use the process of feeding is performed with much less labour, and at the same time more efficiently.

Enumeration of the Plates.

Plates LXXVIII. LXXIX.—Fig. 1 represents a front elevation of the entire mill, showing the form of the framing, and the general disposition of the cylinders or rollers, and of the feeding board, returner and delivering board.

Fig. 2 is an end view of the same, showing the geering by which the rollers are driven.

Fig. 3 is a half-sectional plan taken on the line v—w, in fig. 1.

Fig. 4 is a similar half-section taken on the line x—y.

Fig. 5 is a vertical transverse section, through the centre of the mill, exhibiting the form of section of the framing or standards and the internal construction of the top roller, with its gudgeon and bearings.

Plate LXXX.—Fig. 1 is a longitudinal section of the entire mill, in which the arrangement and dimensions of the rollers and their gudgeons, and the disposition of the feeding board, returner and delivering board, are most distinctly represented.

Fig. 2 is a longitudinal section of one of the lower or feeding and delivering rollers, and of one of the driving pinions.

Fig. 3, a front elevation of the three driving pinions, corresponding in position with their respective rollers, as shown in fig. 1.

Fig. 4, a detached view of one of the stay-bolts for strengthening the standards.

Fig. 5 is a section of part of the feeding board, through the socket of one of its supporting columns.

Fig. 6 is a section of that part of the standard through which passes the screw for adjusting the bearings of the feeding and delivering rollers; and fig. 7 shows a face view of one of these bearings themselves.

General Description.

The crushing rollers consist of three strong cast-iron cylinders A, B, C, mounted between the two massive headstocks or standards D, D, and so disposed that the periphery of the upper roller A is nearly in contact with those of both the others. The rollers are made from 2½ to 3 inches thick, and, to give additional strength, are ribbed in the centre. They are traversed by the strong malleable-iron gudgeons *a*, *b*, *c*, fixed into their respective rollers by keys, in the usual manner, and carrying at one extremity the geering by which the rollers are moved. The gudgeon of the upper roller A is made of considerably greater strength than those of the others, as it has to sustain simultaneously the strain of both. The feeding and delivering rollers B and C have small flanges at their ends, between which the top roller is placed, as shown in fig. 5; these flanges are for the purpose of preventing the pressed canes from working into the mill-bed. Some makers still continue the practice, once universally adopted, of fluting the top roller, in order the better to seize the canes, but it is now very generally abandoned, as it is found, that after working some time, the surface of the rollers becomes sufficiently rough to *bite* the canes effectively, and the fluted rollers have this disadvantage, that the grooves carry round with them a considerable portion of the expressed juice, which is speedily absorbed by the spongy canes, besides causing considerable waste by breaking the canes themselves.

The standards D, D, are securely fixed to the strong cast-iron sole-plate E, which, besides performing this function, is constructed of such a form as to serve as a receptacle for the collection of the expressed juice. For this purpose, that part of the sole-plate marked F, which lies between the two standards, is made to slope downwards from all sides, thus forming a species of trough or cistern, the bottom of which communicates with the gutter *e*, also cast of a piece with the sole-plate, and through which the juice runs off into the proper receptacles. The whole mill rests upon, and is bolted firmly to its foundation G, which, in the example before us, consists of two strong beams of timber, but more generally a stone foundation is preferred. The bolts *f*, *f*, *f*, which serve this purpose, pass through foundation, sole-plate and standards, so that the whole are at once bound together.

The standards D, D, are formed with indentations for the purpose of receiving the bearings H, H, of the feeding and delivering rollers. These bearings consist of a single brass bush for each journal, and their form, as well as the mode of their adjustment, is shown detached from the machine in the detail sheet, figs. 6 and 7. To regulate the distance of the rollers from each other, and to compensate for the wear and tear of the bearings, these latter are so formed as to be capable of being moved to a greater or less distance from the centre of the mill, and for this last purpose the bearings are made of considerable thickness at the points opposite to which the strain is applied. A projecting tongue on the under side of the brass fits into a corresponding groove in the standard, by which means the bearing is guided laterally, and its motion is circumscribed to the required limits by a small projection *d*, cast upon the standard. A small gutter *g*, which is indicated by the dotted lines in the general elevation, is also cast upon the standard round the sole of the bearing, by which the oil applied for its lubrication is prevented from falling into the mill-bed, and is carried round to the outside of the mill. A strong screw *h*, passes through the end of the standard opposite to the centre of each bearing, and works into a nut *i*, sunk into it, for the purpose of adjusting the lower rollers.

The cheeks I, I, of the standards, through which the screws *h*, *h*, pass, are united to, and consolidated with, the main body of the headstocks, by the strong bolts K, K, fixed to the latter by cotters, and secured externally by nuts, after traversing the upper extremities of the cheeks I, I and the cast-iron ferrules *k*, *k*, which serve to fill up the intermediate space. (See the detail fig. 4.)

The axis *a*, of the upper roller revolves in the brass bearings L, L, which consist of double brass bushes fitted into the upper portion of the standards D, D. They are surmounted by the massive caps or covers M, M, which are retained in their places by strong bolts N, N, traversing the whole height of the standards, and secured under the sole-plate by cotters. These bolts serve likewise, by means of the nuts *n*, *n*, to regulate the pressure to which it may be thought expedient to subject the upper roller.

Between the lower rollers is placed a cast-iron plate O, called the *returner;* it is usually made concave upon its upper surface, and is serrated at the edges to admit of the free flowing of the liquor to the mill-bed. At each extremity it is furnished with projecting tails, which pass through slots in the standards, and are supported by the slips of wood P, P, which may be made of greater or less thickness, according as it is found necessary to elevate or depress the returner. The use of the returner is to direct the canes which have been crushed between the top roller A, and the feeding roller C, so that they may be again subjected to pressure between the former and the delivering roller B.

The three rollers A, B, C, are simultaneously set in motion by the strong spur pinions Q, R, S, fixed by keys upon the extremities of their respective gudgeons and geering together, as shown in the detail fig. 3. The pinion of the upper roller, which communicates motion to the others, is itself set in motion by the driving shaft, through the intervention of a clutch or coupling-box, fitting into the teeth *q*, *q*, *q*, which are cast upon it. To provide for the varying resistance arising from irregular feeding, or from the accidental crossing of the canes, by which accidents the engine is liable to be *brought up* so suddenly as to endanger the breaking of the fly-wheel shaft, it is necessary to make all these connections of unusual size and weight. The best surface speed for the rollers is 3·4 or 3·6 feet per minute.

The feed-board P consists of a flat plate of cast-iron, strengthened by feathers on its under surface. It is set at a considerable inclination, and furnished with sheet-iron sides, and its purpose is to convey the canes regularly and equably from the hands of the feeder to the mill. The feed-board rests upon two cast-iron columns *t*, *t*, fixed by cotters at their lower extremities to the edge of the mill-bed. The detail fig. 5 shows the mode of their attachment to the feed-board.

On several sugar estates a continuous system of feeding has been recently adopted, and might, we think, be generally employed with advantage. This consists of an endless web of cloth carried by two parallel rollers, on which the canes are laid. One of the rollers receives motion from the mill itself, and consequently the cloth progresses regularly, carrying the canes with it, and delivering them to be crushed between the feeding and upper rollers. By this means the canes are all presented to the action of the rollers in a longitudinal direction, and in the most equable and regular manner; whereas, when spread on the hopper by the hands of the negroes, the quantity admitted is sometimes too large and sometimes too small, which has the disadvantage, in the one case, of permitting a portion of the canes to pass between the rollers without receiving the due amount of pressure, and in the other, of unnecessarily straining the mill.

The delivering-board U, by which the crushed canes are withdrawn from the mill after the juice has been expressed, consists, like the feed-board, of a cast-iron table, set at a great angle, and fitted close to the delivering roller B, so as to detach any small portions of the canes that may adhere to it, and might otherwise mix with the liquor. It is made so as to turn upon pivots at the top of the small columns *u*, *u*, which support it.

Action of the Machine.—The action of the sugar mill is so obvious as scarcely to require to be specially noticed. The sugar canes, having been previously cut into short lengths of about three feet, are brought to the mill tied up in small bundles; there the feeder unites them, throws them on the feed-board T, and spreads them so that they may cross each other as little as possible. They are drawn in between the feeding and top rollers A and C, where they are split and slightly pressed; the liquor flows down and is received into the mill-bed F, while the returner O guides the canes between the top and delivering rollers A and B, where they receive the final pressure, and sliding down the delivering-board U, are turned out on the floor of the mill, while the liquor runs back and falls into the mill-bed.

When circumstances will admit of it, it is desirable that the mill should be situated at such an elevation above the rest of the sugar apparatus as to render it unnecessary to raise the juice which flows through the gutter *e*, by pumping, as the contact of the air occasioned by the agitation of the liquor in the pump barrels, tends to throw it into a state of fermentation. In very many cases, however, a pump is attached to the sugar mill, and is worked by suitable geering affixed to the gudgeon *a*, of the upper roller, which in our drawings is shown of sufficient length to effect this purpose if required.

Literal References.

A, the upper roller or cylinder.
a, the gudgeon or shaft of the upper roller, upon which it is fixed by keys.
B, C, the delivering and feeding rollers.
b, *c*, their respective gudgeons.
D, D, the standards or headstocks of the mill.
d, *d*, small projections thereon for guiding the bearings of the rollers B and C.
E, the sole plate, to which the standards D, D, are fitted, and which is also formed into
F, the mill-bed, into which the expressed liquor flows.
e, the gutter for withdrawing the liquor from the mill-bed.
f, *f*, *f*, the holding-down bolts of the mill.
G, G, strong beams, forming the foundation of the mill.
g, *g*, gutters for withdrawing the superfluous oil from the bearings of the rollers B and C.
H, H, brass bushes, forming the bearings of the rollers B and C.
h, *h*, regulating screws for the adjustment of the rollers B and C.
i, *i*, their nuts, sunk into the framing.
I, I, the cheeks of the framing, traversed by the screws *h*, *h*.
K, K, cotter-bolts, for strengthening the cheeks I, I.
k, *k*, cast-iron ferrules on the bolts K, K.
L, L, the brass bearings of the top roller.
M, M, the plummer-block covers of the top roller.
N, N, the plummer-block cover bolts, which also regulate the pressure upon the top roller by means of the nuts *n*, *n*.
O, the returner, fixed between the lower rollers, and serrated at each edge.
P, P, small slips of wood for supporting the returner.
Q, a strong spur pinion on the gudgeon of the top roller. On its face are also cast the projections
q, *q*, *q*, engaging with similar projections on the coupling-box of the driving shaft.
R, S, spur pinions on the gudgeons of the lower rollers, geering with the pinion Q.
T, the feed-board.
t, *t*, small columns for supporting the feed-board.
U, the delivering-board, fitted with hinge-joints, to admit of its turning upon
u, *u*, the small columns upon which it is supported.

PLATES LXXXI.—LXXXIII.

WHITELAW AND STIRRAT'S PATENT WATER-MILL.

These Plates contain fully detailed drawings, by Mr. Whitelaw, of the most approved mode of arrangement adopted by him in the erection of this form of hydraulic motor, and likewise of the graphical projection of the arms of the machine.

Plate LXXXI.—This Plate exhibits the general disposition and combination of the parts, and the technical contrivances by which the efficiency and durability of the machine are ensured. Fig. 1 is a side elevation, showing the mode of conveying the water to the machine, the disposition of the bearings of the vertical shaft, and the mode of leading off the power in a horizontal direction. Fig. 2 is a corresponding horizontal plan, on the same scale. The subordinate parts of these drawings are proportional, and therefore no scale has been affixed, the enlargement of one part requiring simply a proportionate enlargement of the others, except in the case of the mill itself, the absolute size of which must vary inversely with the height of the fall. The detached views of the water-joint arrangement, figs. 3—5, are given on a larger scale than that adopted in the main drawings, for the purpose of showing more distinctly the form and construction of the parts.

General Description.

In the general arrangement exemplified in these drawings, and which may be variously modified according to the height of fall, and other local circumstances, the water is supposed to be conveyed from the source by a lead, terminating in the iron *trews* marked E, E. In this is placed the sluice *h*, which can be worked by a small hand-wheel on the upper extremity of the sluice-rod, from the foot-board *i*, laid across and supported on the edges of the trews for that purpose. The sluice-rod is screwed throughout a great part of its length, and passes through the eye or centre of the hand-wheel, which is made deeper than usual on the under side, and screwed internally to fit the rod in the manner of a nut. The prolonged eye has collars formed upon it externally, which are embraced by the brasses of a common pedestal attached to the cross beam of the sluice-frame. The sluice-plate is elevated by turning the hand-wheel, and the water flowing under it, falls through a circular opening at *a*, in the bottom of the trews, and enters the supply-pipe A, A, which is there attached to receive it, and conducts it to the water-mill D, D, in the under side of which is a circular opening, concentric with the axis of rotation, and corresponding in diameter to the piece *c*, in figs. 3 and 5. This piece, which is called the *rising-ring*, is turned truly cylindrical on the part below the flange, and fits water-tight into the conical piece C, C, called the *collar-ring*, by means of a small packing let into a groove cut in it for that purpose (fig. 5). The collar-ring is bored out to the proper diameter, and has a flange round its lower edge, by which it is bolted to the horizontal flange of the re-curved part of the supply-pipe. The general axis of these parts coinciding exactly with the axis of rotation of the machine, and the rising-ring *c*, being lightly pressed upwards against the circular edge of the central opening of the machine, by the three springs *d*, *d*, *d*, (fig. 4), a water-joint is formed with little tendency to leak, and with no appreciable amount of friction. To ensure this, the meeting surfaces are truly turned and ground together; and the tension of the springs is adjusted to a degree just sufficient to maintain the contact, by means of small pinching-screws tapped into the flange of the rising-ring, which is prevented from being carried round with the machine, by a cylindrical pin *e*, passing loosely through a hole in its flange, and entering a corresponding hole bored to a small depth in the collar-ring.

The water-mill, which is keyed upon the lower end of the vertical shaft G, G, consists mainly of two spiral arms, formed according to a rule to be subsequently developed. In order that durability of the brasses of the journals be secured, and friction avoided as far as possible, it is obvious that the weight of the mill itself, and of the shaft to which it is attached, should be just sufficient to counterbalance the hydrostatic pressure acting upon it upwards. This circumstance of the equilibrium of the vertical pressures necessarily induces a modification of the size of the central opening within certain limits; but as this can never be larger than the diameter of the supply-pipe, nor less in area than a certain multiple of the sum of the areas of the orifices, the weight of the machine itself, and of its shaft, must be increased or diminished, in ordinary cases, till the required adjustment is attained.

The water being admitted, as described, to the interior of the machine, flows into the spiral arms *g*, *g*, and escapes by the orifices *f*, *f*, causing them, by its reaction on the opposite surfaces of the arms, to recede in a direction contrary to that of efflux. The machine thus takes a retrograde motion, which being communicated to the vertical shaft G, G, is thereby transferred by the bevil wheels *l*, and *m*, to the horizontal shaft H, leading to the interior of the building in which the power is to be applied. The vertical shaft must have at least two bearings, and in the example given, on account of its length, it is furnished with three. The pedestals of these bearings are usually, as here shown, attached to cast-iron beams *k*, *k*, *k*, passing across the wheel-house; in some cases they are carried on brackets cast on the supply-pipe. The horizontal shaft H has one of its bearings resting on the uppermost of the three transverse beams, and the second in a wall-box placed in an aperture of the wall through which it passes. Beyond this bearing it is provided with a coupling-boss *n*, by which a line of shafting may be engaged with it.

The machine itself is retained in its proper place by the bearings of its shaft. To secure this condition more effectually, the lower bearing is provided with a broad collar, overlapping the flanges of the brasses, so that any vertical pressure not contemplated, and therefore not compensated for in estimating the balance of pressures of the water and of the machine, is received by rubbing surfaces of considerable extent. These provisions ensure a steady motion of the machine; and to provide equally for the stability of the water-joint mechanism, the horizontal part of the supply-pipe A, A, is furnished with a large flange B, parallel to its axis, and serving as a species of sole-plate, by which it is fixed by batt-bolts *b*, *b*, to a foundation of stone-work laid into the bottom of the wheel-pit. By this arrangement the flange supports the weight of the supply-pipe and the column of water contained in it, and as these consist simply of pressure without motion, the foundation is not liable to be deranged. In the fitting of the water-joint, it is, of course, of the utmost importance that the planes of contact of the rising-ring with the central opening of the machine should accurately coincide, otherwise there will be an amount of wear upon the bearings, of loss of power by friction, and of waste of water at the joint, proportional to the amount of inaccuracy.

The expended water escapes from the wheel-pit into the culvert or tail-race by the opening F; and it may be here observed, that this opening and the culvert ought to be sufficiently capacious to allow the water to escape freely, otherwise it may rise so much in the pit as to merge the wheel in back-water, and thereby detract greatly from its power. When, however, the local circumstances are such as to render back-water occasionally unavoidable, the machine is made of a form adapted to work under water, although not with equal efficiency.

Literal References.

A, A, the supply-pipe, receiving the water from the cistern E.
a, a slightly expanded portion of the supply-pipe, at its upper extremity, for the purpose of admitting the water more freely.
B, B, a flange, cast upon the horizontal part of supply-pipe, and secured to the foundation by the batt-bolts *b*, *b*.
C, the collar-ring of the mouthpiece, into which is received the rising-ring *c*, meeting the central opening of the machine, against which it is pressed lightly by the three springs *d*, *d*, *d*. The rising-ring is prevented from being carried round by the machine when in motion, by the cylindrical pin *e*, which attaches together the two rings.
D, D, the water-mill.
f, *f*, the jet-orifices at the extremities of
g, *g*, the arms of the machine.
E, E, the trews or cistern in which the mill-race terminates. It may be formed of wood, as is most frequently the case, or of cast-iron, which is preferable.
h, a sluice placed in the trews, for the regulation of the supply of water to the mill.
i, a foot-board for the convenience of working the sluice.
F, the arched opening by which the water is withdrawn from the wheel-pit.
G, G, the vertical shaft of the machine.
k, *k*, *k*, cast-iron beams built into the wall of the wheel-house for supporting the bearings of the vertical shaft G, G.
l, *m*, a pair of bevil wheels, by which the motion of the mill is transferred from the vertical shaft G, G, to
H, the horizontal shaft, furnished at its opposite extremity with a coupling-boss *n*.

Plates LXXXII., LXXXIII.—These drawings exhibit the graphic details and general scheme of construction of the machine. The parts are proportional, so that the projection may be considered any fractional part of the appropriate size, or even the full size, excepting that when the diameter of the machine is very small, the arms are more usually formed of sheet-copper and the centre of brass, instead of being, as in this example, of sheet-iron and cast-iron. Machines of this description appear to admit of an unlimited variety of dimensions. There are at present in operation water-mills varying from 9 inches to 14 feet diameter, between the centres of the jet orifices, and it is not to be presumed that the practical limits are reached even by the larger of these dimensions.

The method of laying down the projection of the machine is as follows:—Supposing the height of fall and quantity of water to be known, the size of the orifices of efflux is calculated from these data, and from the width of orifice the diameter of the machine is determined, being usually taken at 27 times that unit. The area of the orifices, then, being known, the diameter of the central opening is, by an empirical rule, to be taken equal to ten times the width of one of them; and the height of the fall being given, the hydrostatic pressure is to be calculated, so as to determine the weight to be given to the machine; for as before observed, the vertical pressures must neutralize each other, and allow the machine to revolve, as it were, on a pivot of water. The circles defining the diameter of the machine and of the central opening are, accordingly, drawn in conformity with these data. By the diameter of the machine must be understood the measurement between the vertical planes dividing the width of the orifices equally; and by the diameter of the central opening, the clear width of water-way, as marked by the dotted line in the sectional figure on the right.

To obtain the form of the arms, let the circumference of the diametrical circle be divided into four equal parts; and let three of these quadrants be divided into 27 equal parts; and finally let one of the radii drawn to these points of division be divided into the same number of parts. Through these latter points let circles be described from the common centre of the machine.[*] On one side of the divided radius, lay off three divisions on the diametrical circle; this will be the length of that part constituting more especially the *jet;* and on the opposite side mark off the points of intersection of the successive radii with the concentric circles described from the division of the first radius, taking at each step a point one place nearer to the centre. A curve line passing through all the points thus obtained will define the axis, or centre line of the arm; and for that portion lying on the other side of the divided radius, the diametrical circle itself is to be taken as the centre line. The curve of the arm being thus found, a number of points in the curve lines forming its sides may be determined by taking in the compasses a measure equal to 4½ times the width of the orifice of efflux, and setting off that distance successively from the points of intersection defining the curve of the centre of the arm, upon the circular arcs produced on each side of these points; the divisions thus marked off are points in the two sides of the arm, and being joined by tracing curves through them in the same manner that the axial curve was described, will define the width of the arm from its base to the base of the jet. The sides of this last are described from the centre through an arc equal to twice the width of the orifice on the inside, and three times on the outside; these arcs being joined to the curves forming the spiral sides of the arm, by arcs of mean curvature, the projection of the arm will be complete. And to this the second arm must be made exactly to correspond, either by projecting it in the same manner, or by tracing it by means of a *template* made to the form of that already drawn.

The form of the arms being thus defined, the other details of construction, thickness of metal, &c., must be drawn as exemplified in the sections, from which it will be observed that the machine is made in three parts, besides their subordinate appendages; the central portion, which is of cast-iron, being joined by bolts to the two arms, which are formed of sheet-iron. The depth of the arms, which is uniform throughout, is usually made equal to four times the width of the orifice, as shown in the drawings; but these proportions are susceptible of modification in practice, without detracting from the efficiency of the machine. The arms may also be increased in width, and with advantage; but it is especially requisite that the water be projected in planes perpendicular to the radius of the machine, and that the jet-orifices be so formed, that when the machine is in motion at its proper velocity, they shall discharge at least 95 per cent. of the water due to the head pressure.

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[*] To avoid confusion, these circles are not shown complete in the drawings, but it is necessary to describe them fully in actually laying down the scheme of the machine; it is also necessary to extend the radii to the points of division of the diametrical circle, although they are shown here terminating in the spiral axis of the arm. The object is to find those points in the axis; it is therefore only after this has been accomplished that these extensions can be dispensed with.

These conditions will be found fully discussed in the article on water-power.

PLATES LXXXIV.—LXXXVII.

FOURNEYRON'S TURBINE AT ST. BLAZIEN.

This species of hydraulic machine has attained a high repute on the Continent, although it is still little known in this country. These Plates, indeed, bring it for the first time under the notice of the English engineer in a complete form; and they also exhibit the most notable example of the machine hitherto erected, namely, that of St. Blazien, in the Schwartzwald of Baden, constructed for a fall of 351 feet, from designs and under the immediate inspection of M. Fourneyron the inventor. It is employed in impelling the machinery of a spinning factory of 8000 throstle spindles, with the necessary preparing apparatus. Allowing 120 spindles to a horse-power, it must therefore be working to nearly 67 horse-power, which, at 75 per cent. of the total power due to the water, indicates an expenditure of 134 cubic feet of water per minute. This supply of water is collected from the streamlets on the neighbouring hill, at more than three-fourths of a league distance from St. Blazien, and is received into a reservoir at the height of 351 feet above the machine, whence it is conveyed in a pipe of 1200 feet in length.

Enumeration of the Plates.

Plates LXXXIV., LXXXV.—The principal drawing in these Plates, marked fig. 1, represents a side elevation of the framing and geering of the machine. It conveys a general idea of the external appearance and character of the construction, in which we perceive a degree of elegance and adaptation worthy of this masterpiece of the distinguished mechanician whose name it bears.

Fig. 2 is a half vertical section, on an enlarged scale, of the turbine and crown, showing the mode in which these are fixed to their respective centres, and exhibiting distinctly the manner in which the sluice cylinder operates.

Fig. 3 is a quarter plan of the same parts as the above, with the sluice cylinder and top plate of the turbine removed, in order to exhibit the form and relative disposition of the partition-plates of the turbine, and the direction-plates of the crown.

Fig. 4 is a plan of the sluice geering; and

Fig. 5, a section of one of the sluice-rod pinions.

Plates LXXXVI., LXXXVII.—Fig. 1 represents a general vertical section of the entire machine, shewing its internal construction, and likewise exhibiting very clearly the mode in which the power is transmitted to the main driving shaft of the factory, as well as the arrangements for leading off the water after it has performed its appointed function.

Fig. 2 combines two sectional plans of the machine, taken upon two distinct horizontal planes, indicated by dotted lines in the sectional elevation. That to the left of the centre-line is a plan taken on the line P Q, in fig. 1; while that to the right is taken upon the line R S, at the level of the upper edges of the partition-plates, and also represents clearly the arrangement of the great geering of the mill.

General Description.

The turbine, properly so called, consists of a plate of cast-iron *e*, flat round the margin, and dished towards the middle, in which is a deep eye, bored and fitted upon the turned and slightly conical boss of the central spindle *o*, as shown in fig. 2. The turbine is fixed and retained in its place on the vertical spindle, by a strong nut put on below it, and screwed firmly against the margin of the eye. It is further prevented from being turned round upon the spindle, by a sunk feather or key in the boss. The flat margin of the plate is turned to a true surface, to receive an annular piece which is similarly turned on both surfaces: the under surface to lie truly on the turbine-plate, and the upper to form a smooth channel for the water. This ring is divided off into 36 equal compartments by the curvilinear diaphragms *c*, *c*, the curvature of which is determined according to certain mathematical rules developed in our Treatise on Water Power.* Each of the compartments thus formed opens into a corresponding division of the internal part *f*, called the *crown*, and which at its circumference is divided off into compartments by the direction-plates *d*, *d*, the curvature of which is likewise determined by rules which are considered essential to the proper action of the machine. These diaphragms are fixed vertically, and, in this example of the machine, their edges are turned over at right angles to the planes of the plates, in the manner of angle-irons, the horizontal part forming a species of flange. In the case of the turbine diaphragms, *c*, *c*, both of the edges are thus laid over, and are let into grooves cut into the sole and cover plates, flush with their respective surfaces. The flanges being thus fitted, they are riveted to the plates into which they are inserted, so that while they are retained in their relative positions by the two discs, they serve to bind these effectually together, and to divide off the space between them into an equal number of horizontal compartments, defined laterally by the convex surface of the diaphragm on the one side, and by the concave surface of the contiguous diaphragm on the side opposite. To complete the turbine, strictly so named, the under plate of this compound annulus is fixed on the main sole *e*, by riveting, the rivets being counter-sunk in the upper surface, that they may offer no obstruction to the passage of the water.

The direction-curves *d*, *d*, are fixed into an annular disc in the same manner, but have no cover-plate, as in the case of the partition-curves of the turbine. The sole-plate upon which these diaphragms are carried is recessed into the crown *f*, flush with the general surface, and riveted in its place. It is also to be observed that the height of the direction-plates is considerably greater than the depth of the water-space of the turbine, and that they rise into a recess formed in the metal of the sluice cylinder *s*, which serves to them instead of a fixed cover, but admits of being raised and depressed at pleasure by means of a system of geering represented in fig. 4. The position of this cylinder, by thus regulating the depth of passage through which the water escapes into the turbine, serves to adjust the capacity of the machine, within certain limits, to the disposable supply of water, and consequently to the power which may at any time be requisite for the work to be performed.

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* See page 240—Essays.

The crown *f*, is a strong disc of cast-iron immovably fixed upon the lower end of a vertical tube *j*, inclosing the spindle of the machine. The crown occupies the dished cavity of the turbine-plate, within the inside diameter of the rim, and supports at its circumference the annular sole of the guide curves *d*, *d*, as before described. The object which this part is intended to serve is to support the vertical pressure due to the height of the head column under which the machine is intended to act, and which, but for the interposition of the crown-plate, would press with its entire weight upon the turbine-plate, and greatly increase the friction and wear of the footstep of the spindle. The tube *j*, ascends through the whole height of the water-cistern *b*, as shown in the general vertical section, and passing through an eye bored in the centre of the cover-plate, is secured above by a strong cast-iron nut. It is prevented from turning round by a sunk key, and from rising, or being drawn up by the nut, by a shoulder which fits against the under margin of the central boss of the cover. The joint is thus fitted accurately to the proper height of the crown, and is perfectly water-tight.

This tube, besides supporting the weight of the crown, and the pressure of the superincumbent water, serves also to allow the spindle *o*, of the machine, to be carried through the water-cistern without the necessity of employing stuffing-boxes at top and bottom. The spindle passes through the interior of the tube, and is fitted accurately to it at the upper and lower extremities, but at no other points throughout the entire length. These short bearing surfaces are sufficient to prevent all lateral movement or tremor; and any enlargement of them would only tend to increase the amount of friction and difficulty of construction. The small quantity of water which falls into the cavity of the turbine-plate between the circumference of the crown and the inner edge of the rim of the turbine, is allowed to escape through apertures formed in the plate *e*, for that purpose, and cannot, therefore, so accumulate as to ascend into the tube-casing *j*.

The water-cistern *b*, is a strong cast-iron cylindrical vessel, with a broad flange at the upper end, by which it is suspended in a circular opening, formed in the centre of the entablature frame. It is contracted towards the base into a cylinder of smaller diameter, and is there bored to receive the end of the sluice cylinder *s*, which is fitted into it, and rendered water-tight by means of a small packing of hemp-band inserted into a circular recess cut in the inner surface of that part of the cistern. The cistern is provided with a strong cover *g*, cast with radiating feathers on the under side, and with a circular web, or vertical flange, which is turned to fit accurately into the mouth of the cistern, while the overlap flange is made to fit with equal accuracy to the faced surface of the cistern flange. The bearing surfaces being washed with a solution of sal-ammoniac, and the cover made fast in its place by bolts tapped into the metal of the cistern, a water-tight joint is thus formed, and the relative accuracy of the parts is at once secured.

The supply-pipe *a*, terminates in the water-cistern *b*. This pipe ascends obliquely to the height of the upper reservoir or dam, through a distance of 1200 feet, which great length accounts for its comparatively large internal diameter of 18 inches. It is formed in 300 lengths, of four feet each, diminishing in strength upwards from "five to four-tenths of an inch" thickness of metal. To withstand the lateral shock of the water falling from the great height of 351 feet, in a column of 18 inches diameter, the lower end of the cistern is supported by six oblique stay-rods *x*, *x*, attached to the columns of the framing, and abutting against the contracted part of the cistern, into which the sluice cylinder is fitted. The body of the sluice, as before stated, is simply a cylinder of cast-iron, accurately turned and fitted water-tight into the lower end of the cistern. Its lower edge is swelled inwards to admit of an exterior recess being formed, into which the diaphragms *d*, *d*, of the crown, are received. This recess is surrounded externally by a cylinder of plate-iron *l*, to prevent the escape of the water which rises into it between the diaphragms, and which, but for this provision, would be projected entirely over the rim of the turbine, in obedience to the pressure within.

The sluice cylinder is embraced towards its lower extremity by a strong ring cast with arms to receive the lower ends of the four rods *y*, *y*. The ring is made fast in its place by four bolts, for which suitable projections are provided upon the exterior of the cylinder; the rods *y*, *y*, are also fixed in the arms of the collar by nuts above and below, for the purpose of adjustment. The rods are also screwed at their upper extremities, where they enter the internally screwed eyes of the four spur pinions *k*, *k*, *k*, *k*, shown in the plan, fig. 4; one of the pinions is also shown in section at fig. 5. These pinions geer with a large central wheel *m*, *m*, which is free to revolve upon a hollow centre cast on the cover of the water cistern. This wheel again geers with a small intermediate pinion carried on an axis attached to the circular frame *r*, *r*, in which are the bearings of the four pinions *k*, *k*, *k*, *k*, and which is made fast by stud-bolts to the centre formed by the intersection of the diagonal rails of the entablature. This small pinion geers with a worm or endless screw on the end of the long spindle *p*, which passes to the interior of the factory, where it can be worked at pleasure by a hand-wheel placed upon its extremity. This worm being put in action, its motion is transferred by the pinion with which it geers to the large central wheel *m*, *m*, and thence to the four pinions *k*, *k*, *k*, *k*. But these pinions acting as fixed nuts upon the rods *y*, *y*, the latter will be elevated or depressed through a space proportioned to the common arc described by their respective pinions. The sluice will thus be elevated when the rod *p*, is turned from left to right, and *vice versa*, thus enlarging or contracting the sluice opening, and thereby allowing a greater or less supply of water to pass into the turbine, according to the amount of power required, within the limits of the disposable supply of water.

The lower part of the framing of the machine is entirely enclosed, to form a cistern for the reception of the water which issues from the machine, and to prevent the spray from being thrown upon the geering. The expended water is drawn off by a covered gutter, into the culvert or tail-race L, situated under the floor. The wheel-house is thus kept perfectly dry; no water whatever can be observed, and for any thing which the casual observer can discern to the contrary, the machinery in the interior of the framing may be impelled by any other agency than that actually employed.

The spindle of the machine is carried at its lower end on a steel pivot sunk into it, and which works in a footstep, so as to admit of a small amount of vertical adjustment by means of the cotter and gib *h*. At its upper extremity, it works in a bearing in the end of a strong arched bracket bolted to the entablature of the framing—an arrangement scarcely in accordance with the generally substantial character of the other parts. On the upper end of the spindle, is fixed the strong spur pinion *t t*, which geers with the large mortise wheel *u u*, surmounting the second vertical shaft *q q*. On the lower end of this shaft is fixed a bevil wheel *v v*, which in like manner geers with a large mortise wheel *w w*, on the horizontal shaft *n*. The foot of the second vertical shaft is carried in a step supported by the bridge *i i*, which is carried on the sole of the frame, and its upper end works in a pedestal bearing resting on the entablature.

Action of the Machine.—From what has been above stated in reference to the particular functions of the several parts, it is easy to understand the general mode of action of the machine. The sluice being raised to a given height, determined by experience, the water passes through the compartments formed by the direction-curves *d*, *d*, on the margin of the crown, at a determinate angle to the plane of the diaphragms of the turbine, (with a velocity proportional to the square root of the height of the impelling head, and therefore with an impulsive force directly as the height of that head), impinging against the inversely curved or hollow faces of the turbine plates. These recede from the impulse, and from their connection with the central shaft *o*, a centrifugal force is imparted to the water, which causes it to recede from the centre of motion, and escape outwards, with a force and velocity determined by the angular velocity of rotation. But in consequence of the increase of pressure thus produced in the compartments of the turbine by the centrifugal force, and the consequent greater velocity of escape at the circumference, a certain amount of re-action is added to the impulsive force of the fluid, giving thereby an increase of power. But from this is to be deducted the power consumed in giving rotatory motion to the volume of water expended by the machine, and which will be found to exceed the increment due to the centrifugal action. The discussion of these particulars cannot, however, be entered upon in this place, but will be fully treated in that part of the present work which is devoted to the investigation of the principles on which the action of hydraulic motors depend.

Literal References.

a, the supply-pipe bolted to
b, the water-cistern or body of the machine.
c, *c*, the curved diaphragms of the turbine.
d, *d*, the direction-plates of the crown.
e, the turbine-plate, or main sole.
f, the crown, forming in effect a bottom to the water-cistern.
g, the cover of the water-cistern.
h, cotter and gib for adjusting the footstep of the central spindle of the machine.
i, *i*, the foot-bridge on which the step of the intermediate vertical spindle is carried.
j, the casing of the main spindle *o*, serving also to sustain the crown *f*.
l, the outer casing of the direction-plates.
k, *k*, *k*, *k*, four spur pinions which act as nuts upon the sluice-rods *y*, *y*, and geer with
m m, a central spur-wheel giving motion simultaneously to the pinions.
n, the horizontal shaft by which the motion is conveyed to the interior of the factory.
o, the spindle of the turbine.
q, the intermediate, or second motion vertical shaft.
p, the worm-spindle by which the sluice is worked from the interior of the factory. It is set in an oblique position on account of the worm being made to work into a spur pinion, the angle of the spindle compensating for that of the thread of the screw, which is thereby brought vertically to the edge of the wheel.
r, the cover-frame of the sluice geering, in which are formed the upper bearings of the pinions *k*, *k*, *k*, *k*.
s, the sluice-cylinder.
t t, the first motion pinion on the top of the spindle *o*.
u u, a large mortise wheel on the top of the second shaft, geering with the pinion *t t*.
v v, a bevil wheel on the bottom of the second shaft.
w w, a large bevil mortise wheel on the horizontal shaft *n*, geering with the wheel *v v*.
y, *y*, sluice-rods, (four in number), attached to a ring on the sluice cylinder, and passing through the screwed pinions *k*, *k*, *k*, *k*.
L, L, the culvert or tail-race.
P Q, and R S: these letters indicate the lines on which the two sections combined in fig. 2, Plate LXXXVI. are taken.

PLATES LXXXVIII.—XCI.

OVERSHOT WATER-WHEEL. BY JAMES SMITH OF DEANSTON.

The two species of hydraulic machines which form the subjects of the Plates immediately preceding the present, exhibit the more modern application of the power of water to the propulsion of machinery, and although, in cheapness of construction and in their applicability to certain peculiar circumstances, they present many advantages over the more established methods, it is not to be presumed that they will ever entirely supersede the use of the various kinds of vertical water-wheels, *overshot*, *undershot*, and *breast*. We have, therefore, in our collection of engravings, given specimens of each of these classes of prime movers, and we have been careful to select the best and most modern examples within our reach.

The Plates now before us present general and detailed views of an overshot water-wheel, which may not only with confidence be pronounced a most perfect specimen of its class, but which, moreover, when seen in motion, affords a more striking and impressive evidence of the power of the mechanician to make the stubborn materials of his art bend to his will, than is exhibited, perhaps, by any combination of machinery at present in existence. Its vast dimensions, and the silent majesty, yet apparently resistless force, of its motion, impress upon the spectator a feeling approaching to the sublime, and, should he possess the practised eye of the mechanic, he will not fail, upon a minuter examination, to be equally struck with the admirable combination of lightness and strength exhibited in the details of its construction. The wheel itself, together with its accompanying heavy geering, governor and sluice geering, is the workmanship of James Smith, Esq. of Deanston, a gentleman whose reputation as a scientific mechanician has been long established in many department of mechanics, and is admirably sustained by the taste and skill he has evinced in planning and executing this gigantic piece of mechanism.

The factory to the machinery of which this wheel gives motion, is the property of the Shaws Water Cotton Spinning Co. of Greenock, and is situated on the banks of the Shaws Water, which is the name given to an artificial accumulation of water on the hills above Greenock, and which descends to the sea at that port, in a series of thirty-three falls, affording a very abundant source of power, which has been made extensively available for various manufacturing purposes. The height of the fall at the point where this wheel is situated, is about sixty-four feet, and the quantity of water which it receives is 2100 cubic feet per minute,* which data, allowing that the wheel gives seventy-five per cent. of the total power of the water, indicate a power of 192 horses. The factory contains 12,216 mule-spindles, and 13,544 throstle-spindles; and assuming, at a rough average, that one horse-power will drive 270 mule-spindles, and 120 throstle-spindles, with the necessary preparation, and allowing one-fourth of the whole power to be consumed in overcoming the friction and inertia of the wheel and geering, the result of this calculation will be found to agree nearly with the power obtained theoretically as above indicated.

Enumeration of the Plates.

Plates LXXXVIII., LXXXIX. This sheet contains, first, a side elevation of the whole wheel, one half shown complete, exhibiting an external view of the centre, arms, and shrouding, while the other half is a section taken on a vertical line passing through the middle of its breadth, for the purpose of showing the form and disposition of the buckets. In this view the mill-race, sluice and sluice-governor are also shown in their true position in relation to the wheel, as likewise the heavy geering by which the motion of the wheel is multiplied and transferred to the interior of the factory.

The figure towards the right of the sheet combines an external end elevation of the wheel with a vertical section through the centre. In the first is seen the external appearance of the buckets, and of the large internal spur-wheel by which the power is communicated to the heavy geering of the mill; and in the second is exhibited the form of cross section of the centre and of the shrouding, (the line of section of the upper half being taken through one of the square, while that of the under half is taken through one of the diagonal arms). This view also shows distinctly the form of the shaft on which the wheel revolves and the arrangement for its lubrication.

Plate XC. This Plate contains general and detailed views of the sluice-governor.

Fig. 1. A front elevation of the governor, showing the whole general arrangement of this very complete and elegant piece of mechanism.

Fig. 2. A general end elevation as seen from the side towards the wheel.

Fig. 3. A partial end view as seen from the opposite side.

Figs. 4—13. These figures exhibit detailed views, on an enlarged scale, of the principal parts of the governor, detached from the machine itself.

Plate XCI. This plate is occupied with the constructive details of the water-wheel itself, and of its sluice and sluice-geering.

Fig. 1. A side elevation of a portion of the shrouding and of the great internal spur-wheel, showing the mode in which the segments of each are joined together, and also the form and dimensions of the arms.

Figs. 2—9. These figures are introduced for the farther illustration of the mode in which the parts of the wheel are put together.

Figs. 10 and 11. A vertical section and end elevation of part of the buckets, showing the mode of their attachment to the sole and to the shrouding, also the number and form of the stays between the buckets.

Figs. 12 and 13. A plan and elevation of the plummer-block in which the main shaft of the wheel revolves.

Fig. 14. A view of the eccentric for altering the position of the wheel should circumstances render it necessary.

Figs. 15 and 16. A longitudinal section, and face view of the sluice and geering by which it is worked. In these views the construction of the cistern in which the mill-race terminates is also indicated.

General Description.

The Shrouding.—The rim or *shrouding* A, A of this wheel is composed of thirty-two cast-iron segments for each side, which are fitted and jointed together by means of half-lap joints in the manner indicated in figs. 1, 2, 3, and 4, of the detail sheet. Each segment is furnished with a flange *a*, projecting internally, and abuts against the contiguous segment at *b*, in such a manner as to give no external indication that the wheel is composed of separate parts. To

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* About two years ago, an accident having occurred to the great reservoir, its contents had to be run out; consequently, the quantity supplied for machinery was for a time curtailed. To meet this deficiency, the proprietors of this mill took another fall adjoining their works, of 22 feet, on which they erected a water-mill, on Whitelaw and Stirrat's construction. The water, when passed through this wheel or mill, is led on to the great wheel, and the power of both united. The quantity of water which can now be supplied is equal to 2400 cubic feet per minute, for 12 hours per day, the average rent of which does not exceed £3 per horse-power.

give greater firmness to the joints, they are made to bear throughout their entire surface, and not merely upon fitting strips,[*] as is usual in work of this kind; and, by an ingenious combination of machinery which it is not necessary here to describe, Mr. Smith has been enabled to finish the whole of the joints in this wheel without having recourse to the tedious and expensive process of chipping and filing. The length of each of the segments of the shrouding is 6 feet 10 inches, thus making a total circumference of about 220 feet, which is equivalent to a diameter of 70 feet 2 inches. The breadth of the shrouding is 17¼ inches.

The Buckets and Sole-plate.—The buckets B, B, of which there are 160, being 5 to each segment, are made of sheet-iron, and are constructed of such a form that, while they allow the free admission of the water at the sluice, they shall retain it as long as possible within themselves. Each bucket is capable of containing about 100 gallons of water; but this wheel, like all others of its class, is found to work most advantageously when it revolves at such a speed as to allow of the buckets being only two-thirds full. A few small holes in the bottom of each bucket are found to be sufficient for the purposes of ventilation.

The ends of the curved plates B, B, which form the sides of the buckets, are riveted to narrow projecting shelves or *runners* *c*, *c*, of a corresponding curvature, cast on the inside of the shrouding, and a similar flange *d*, *d*, is formed all around its inner edge, for the purpose of serving as the point of attachment for the sole-plate C, C. The plates B, B, are attached to the sole-plate by strips of angle-iron *e*, *e*, riveted to both, in the manner indicated in fig. 10, of the detail sheet. To give the necessary degree of rigidity to the buckets, they are strengthened by the malleable iron stay-bolts *f*, *f*, *f*, passing through each contiguous pair, and secured to the proper tension by nuts and jam-nuts.

The Arms and Centre.—This wheel is constructed upon what is called the *tension*, or *suspension* principle, or as it is sometimes termed, with *spider-arms*. This mode of construction is applicable where circumstances admit of the power being transferred to the main geering at or near the circumference of the wheel, as thereby the arms have to sustain little strain beyond that necessary for retaining the system of buckets and great spur wheel within their prescribed sphere of action. Accordingly the arms of such wheels may be made of extreme lightness, as compared with the power generated by them. In the wheel now under consideration, the shrouding is braced to the centre by 32 chain-cable iron bars or arms D, D, D, 2¾ inches diameter, and by an equal number of diagonal braces E, E, E, of 2½ inches diameter. The mode in which these arms are fixed into the shrouding is shown in figs. 5, 6 and 9, Plate XCI. The arms are made with strong T-heads, and are sunk into similarly formed recesses, cast in the projecting bosses *g*, *g*, and *h*, *h*, on the inside of the shrouding. It is to be observed that these recesses, and the ends of the arms which fit into them, are made with a little dove-tail, so that, when the arms are drawn up towards the centre, they shall have no tendency to slip out of their places. The projections *g*, *g*, and *h*, *h*, are sloped away very gradually into the internal surface of the shrouding, in order to obviate, as much as possible, the risk of strain to which the metal is exposed by its unequal thickness.

The opposite extremities of the arms are made of a square form and fitted into the square sockets *i*, *i*, and *j*, *j*, formed for their reception in the strong cast-iron centre F. Here they are fixed in their places by gibs and cotters, as shown in the general vertical section, Plate LXXXIX. The sockets *i*, *i*, for the square arms D, D, are, of course, all situated in the plane of the shrouding and centre, but the others *j*, *j*, which are also cast on the centre, and into which the diagonal arms E, E, are inserted, are made to radiate, each in the direction of the points in the *opposite side* of the wheel, at which their respective arms are fixed. The centres F, F are strong cast-iron plates 3½ inches thick, formed with eyes 28 inches diameter, for the reception of the ends of the shafts G. They are strengthened by radiating feathers opposite to each of the square arms, and by semicircular mouldings springing from the roots of the sockets into which they are fixed. The centres have thus, notwithstanding their great size and weight, a light and elegant appearance, which is perfectly in keeping with the characteristic features of the rest of the work.

The shaft G G, which forms the axis of rotation of the wheel, is made of cast-iron; the form of its cross section at the ends and centre is shown in the principal view, Plate LXXXVIII. Each of the centres bears against a shoulder cast upon the shaft, for the purpose of keeping them at their proper distance, and the strain of the diagonal arms tends to sustain the whole structure firm and immovable. Eight keys *k*, *k*, are employed for fixing each centre to its place on the shaft.

The journals of the shaft, which are 18 inches diameter, and 24 inches long, revolve in cast-iron bearings fitted into the plummer-blocks H, H. The strain being altogether in a downward direction, owing to the great weight of the wheel, the usual solid plummer-block cover is dispensed with, and instead of it a false cover is substituted, consisting merely of a thin shell of cast-iron, which affords a convenient arrangement for the lubrication of the journals. An oblong box *l*, is cast upon the cover, and fitted with a lid, movable upon a hinge. Into this box the lubricating material (tallow) is introduced, so as to fill the interior of the cover, and thus present the whole upper surface of the journal to its constant and equable action.

A very simple and ingenious contrivance has been introduced by Mr. Smith, for the purpose of effecting any slight alteration which may at any time be necessary, in the height or position of the wheel. From the enormous weight of the wheel, (117 tons), it would be almost impossible to find tackle or levers sufficiently powerful to move it; it was, therefore, desirable that a portion of its own stupendous power should be applied for this purpose. This is effected by simply fixing a solid eccentric *m*, upon each end of the shaft, and forming a part of the sole-plate *n*, upon which the plummer-block is fixed, into a species of shallow oblong trough, under the eccentric. Into this recess, when it is required to raise the wheel, a block of wood is introduced of sufficient thickness to fill the space between the sole-plate *n*, and the thin side of the eccentric *m*. The water is then admitted to fill the buckets, and by the revolution of the wheel, the opposite side of the eccentric is slowly brought round until the whole mass is raised to the required height.

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[*] There is a slight discrepancy between our drawings and the description of these parts. The drawings are, in this respect, not strictly accurate.

The Foundation.—The *arc* or foundation, on which the axle plummer-blocks rest, together with the arch of the tunnel by which the water is withdrawn from the wheel after having performed its function, is constructed of dressed freestone, the joints being all joggled and filled with cement. The stones forming the arc weigh from one to ten tons each, and the whole consists of 5000 tons of solid masonry, ten feet in thickness. The plummer-blocks of the wheel are elevated upon a stone platform about 5 feet above the general level of the arc, to admit of the sole-plate for the heavy geering being placed at a level the most advantageous for the transference of the power of the wheel. The holding-down bolts, *o*, *o*, *o*, by which the sole-plates are fixed, pass through nearly 20 feet of the solid mason work. They are 2½ inches diameter, and are secured below by strong washers and cotters. The floor of the tail-race is about 16 inches below the level of the lowest point in the circumference of the wheel.

The Heavy Geering and Calculation of Speeds.—To the shrouding, on that side of wheel which is situated towards the factory, is fixed the great internal spur wheel I I. Like the shrouding to which it is attached, it is made in 32 segments, which are bolted to each other by ribs cast on the back of the rim, and to the shrouding by projecting palms fitting into beds cast upon the outside of the shrouding, and accurately prepared to receive and bind them in their places without allowing any play. The technical details of these arrangements will be at once understood by a reference to figs. 1, 7 and 8, in Plate XCI.; observing, however, that all the joints are made to bear throughout their entire surface. This is quite necessary, as, in spite of all the precautions that can be taken, so great is the strain upon these parts, that it is almost impossible to impart sufficient firmness and solidity to them. The dimensions of the spur wheel are calculated so that the pitch line of the teeth should be in the circle which passes through the centre of gravity of the water in the loaded buckets, the *rationale* of which arrangement is sufficiently obvious. In each segment there are 22 teeth; consequently, the whole wheel contains (22 × 32) 704 teeth. The wheel I, geers with a pinion J, of 62 teeth, overhanging its bearing upon the edge of the arc, and upon the same shaft, close to the wall of the wheel-house, is fixed a large spur wheel K, of 208 teeth, geering with another pinion L, of 63 teeth, fast upon the main shaft which passes into the interior of the factory, and gives motion to its varied and complicated machinery. The plummer-blocks for the bearings of these shafts are fixed to one massive cast-iron sole-plate, to ensure greater compactness and steadiness of motion, and the holding-down bolts *o*, *o*, include within their grasp a very large mass of the solid masonry which forms the foundation.

The water-wheel performs 1⅓ revolutions per minute; consequently, the velocity of the wheels J and K will be $\frac{704}{62} \times 1{\cdot}333 - 15{\cdot}1375$ revolutions per minute, and that of the main shaft into the mill, $\frac{208}{63} \times 15{\cdot}1375 = 49{\cdot}9537$, or nearly 50 revolutions per minute. And yet this great increase of velocity from 1⅓ to 50, and the transmission of the enormous power of nearly 200 horses, is accomplished within such small limits without producing the slightest noise or vibration in any of the parts; a sufficient proof of the judiciousness of the arrangement, and of the excellence of the workmanship.

The Sluice and Sluice Geering.—The trews or cistern M M, which conducts the water to the wheel, is made of iron plates riveted together, as shown in the detail views, figs. 15 and 16, the upper edges being strengthened by semi-circular mouldings of malleable iron on the outside, and by angle irons running along the inside, also riveted to the plates. It is 3 feet in depth and extends across the whole breadth of the wheel. The point in the circumference of the wheel at which the water is laid on is about 6½ feet from the top of its diameter; thus forming an angle, of about 130°, between the surface of the water and the radius of the wheel. The trews is supported by two strong cast-iron beams, extending across the whole width of the wheel-house (33 feet), and on the same level a platform has been constructed, for the convenience of working the sluice, and otherwise regulating the motion of the wheel. The sluice-face *p*, is of cast-iron riveted to the plates of the cistern, and fitted to it at such an angle as to coincide with the circumference of the wheel throughout its whole depth. The water passes from the trews into the buckets through two large apertures in the sluice-face, and the supply is regulated by means of the cast-iron plate or cover *q*, extending across the whole breadth of the sluice-face, and fitted to it so as to be perfectly water-tight. The sluice cover is worked by two racks *r*, *r*, fixed near the two ends, and supported at the back by two small friction rollers. These geer with the pinions *s*, *s*, which are fast upon a wrought-iron shaft extending across the cistern, and having its bearings in two cast-iron plummer-blocks bolted to its sides. A spur-wheel *t t*, 2 feet 6 inches diameter, is fixed upon the end of the sluice-shaft, and geers with a small pinion *u*, working upon an intermediate stud bolted to the side of the cistern; this pinion is fast to a wheel *v*, 15 inches diameter, working upon the same axis, and receiving motion from another pinion *w*, upon the shaft *x*, which communicates with the governor N. This system of geering is found to be necessary in order that the action of the governor upon the sluice may be very slow and gradual, as well as powerful; were it otherwise, the wheel would be liable to incessant oscillations and changes of velocity.

The governor N, is driven by a slender vertical shaft *y*, set in motion by a pair of small bevil wheels *z*, the horizontal shaft communicating, by means of a system of light geering, with the main shaft into the mill.

Description of the Governor.

The governor of this water-wheel, which, in our general views, we have designated by the single letter N, is such a complete and beautiful piece of mechanism, and its various arrangements are at once so ingenious and so effective, that we have esteemed it worthy of separate representation on an enlarged scale, as an excellent specimen of this indispensable adjunct to prime movers of all kinds. We have accordingly given, in Plate XC., both general and detailed views of the governor, which, as well as the wheel to which it is attached, has been produced in the workshops of Mr. Smith of Deanston.

The framing A, A, to which the main part of the governor is attached, is a cast-iron bracket bolted to the wall of the wheel-house, near the level of the cistern by which the water is led on to the wheel. The upper surface of this bracket forms a broad table, on which are bolted the two standards B, B, provided with bearings *a*, *a*, for the horizontal shaft by which the action of the governor is transmitted to the sluice-shaft; and formed at their point of junction into a central socket *b*, through which the vertical spindle of the governor passes, working in a brass bush fitted into it. The construction of this part will be at once understood by reference to fig. 4. The upper ends of the standards B, B, are cast in a semi-cylindrical form; these are carefully fitted together, and connected immovably by a small wrought-iron hoop *c*, accurately turned to fit them and shrunk on; the whole forming a very elegant and appropriate frame-work for the governor. On the front and back of the standards B, B, are bolted the two light cast-iron cross-stays C, C, and these are connected together by the malleable iron bridge D, in the centre of which a bushed socket supports the lower end of the governor spindle.

The vertical shaft *d*, *d*, by which the governor is driven, and which receives its motion in the manner adverted to at the close of the description of the wheel, (where it is indicated by the letter *y*), has its upper bearing at E, in a cross piece bolted to the bracket A, and carries at its extremity a small bevil wheel *e*, geering into two others *f*, and *g*, of the same dimensions, running loose upon the cross shaft of the governor. By this disposition, it is obvious that if we suppose the wheels *f*, and *g*, to be fast to their common axis, they would each impart to it a motion in a direction contrary to that of the other; but being loose, they serve in the first instance, merely to transmit the motion of the wheel *e*, to the remaining bevil wheel *h*, which completes the system, being fixed to the lower extremity of the governor spindle *i*. At its upper end is a small brass socket *j*, to which the legs *k*, *k*, of the governor balls G, G, are jointed. These rods and the links *l*, *l*, which are attached to them about the middle of their length, are guided and constrained to revolve with the governor spindle *i*, by a double radial arm F, fixed to the latter, and slotted to receive the rods *k* and *l*. The lower ends of the links *l*, *l*, are jointed to a cylindrical brass slide *m*, fitted to move up and down upon the governor spindle as the balls diverge and converge in obedience to the action of the centrifugal force. The lower portion of the slide *m*, is formed into an eccentric *n*, (see fig. 6), which revolves between the two small steel catches *o*, *o*, inserted into the ends of a double bent lever H, (fig. 5), and so adjusted that the eccentric shall come into contact either with the upper or lower catch, according as the slide *m*, is elevated or depressed by the action of the governor balls. The piece H is fixed to the end of an upright lever I, having its fulcrum in the small vertical spindle J, working in bearings at the back of the governor, and which carries a forked lever *p*, (fig. 13), adapted to move the double clutch-box K longitudinally upon its axis in the cross shaft *q*, of the governor, to which, however, it is attached by a sunk feather, so that it cannot revolve without carrying the shaft with it. The teeth of the coupling-box K are fitted to engage with similar projections cast upon the insides of the small bevil wheels *f*, and *g*, which, as was formerly remarked, run loose upon the same axis *q*. The whole arrangement of these parts is shown in the vertical section, (fig. 7).

The cross or horizontal shaft *q*, is made in two pieces, which are connected together by a sliding coupling-box L, (figs. 9 and 10), for the convenience of disengaging the governor from its connection with the sluice-shaft, when it is required to stop or start the wheel. The coupling-box is worked by means of a forked cast-iron lever *r*, embracing the loose coupling, and having its centre of motion in the bracket M, bolted to the upper surface of the main bracket A. A slender rod *s*, formed into a handle at its extremity, serves to bring this lever within the command of the attendant to whom the charge of the wheel is entrusted. Here is also situated the bracket O, carrying the bearing for the outer extremity of the shaft *q*, on which is fixed a hand-wheel P, for the convenience of working the sluice by hand

The shaft *q*, carries an endless screw or worm N, formed of cast-iron, and working into the worm-wheel Q, fast upon the shaft *x*, leading to the sluice geering, which we have described in treating of the sluice. We may here call attention to the peculiar adaptation of this arrangement to the purpose to which it is here applied, as it not only furnishes at once the requisite power and diminution of speed, but, from a well-known property of the endless screw, prevents any strain from being transmitted through it to the governor. The worm-wheel may be disengaged from the endless screw when required, by means of the lever S, the action of which will be better understood by the sequel.

We shall now proceed to describe a very ingenious contrivance which has been applied by Mr. Smith to this governor, for the purpose of causing the wheel itself to arrest its own motion, in case of a *break-down*, or any accident tending to accelerate its velocity beyond the prescribed limits. A pair of meter wheels *t*, and *u*, of the same dimensions as those employed in the governor itself, transfer the motion of the vertical shaft *d*, to the short horizontal shaft *v*, situated under the governor, and having its bearings in two small plummer-blocks bolted to the brackets T and U. A similar pair of meter wheels *w*, and *y*, are employed to change once more the direction of the motion, and to give continuous rotation to another short horizontal shaft z, at right angles to the former, revolving in bearings bolted to the bracket U. The shaft z, is situated exactly under the shaft x, leading to the sluice-geering, and carries a species of drum or pulley V, covered with thick leather to give greater friction, and revolving directly under the worm-wheel Q, (see figs. 1 and 3). The system of wheels t, u, w, y, is so arranged that if we suppose the worm-wheel Q to come into contact with the drum V, and to press upon it with its own weight and that of its shaft, the motion thereby communicated to the sluice-geering would quickly shut off the water from the wheel.

To satisfy this condition it is necessary that the worm-wheel Q should be capable of being disengaged from the endless screw N, and of falling into contact with the drum V. For this purpose the bush in which the end of the shaft x, works, is fitted to slide up and down in the cast-iron carriage R, bolted to the wall of the wheel-house, and is jointed to the end of a long bent lever X, having its centre of motion in the side of the carriage R. This lever is made of such a length that its opposite end is exactly in the same vertical plane with the centre of the governor spindle i; and to this end is jointed the upright rod W, having two catches at its upper extremity, which is pressed outwardly from the wall by a slight spring, (see fig. 2). When the worm-wheel Q is in geer with the endless screw N, and every part of the machinery, consequently, is in full operation, one of these catches bears against the under edge of the guide which supports the rod W in its vertical position; but should the governor balls, by any sudden increase of velocity, diverge beyond their proper limits, they strike the upper catch, and the other being thereby released, the worm-wheel and its shaft fall by their own weight upon the rim of the drum V, and the sluice is almost immediately closed.

In case of any accident occurring in any of the flats of the mill, which renders it necessary that the machinery should be immediately stopped, without the delay incurred by conveying the intelligence to the wheel-keeper, a very simple but effectual contrivance has been provided, in connection with the parts we have just been describing. A long lever Y, Y, having its centre of motion in a small bracket fixed to the wall of the wheel-house, bears against the rod W, and to its opposite end is attached a wire, branching off to all the flats in the manner of a common bell-wire; on pulling any of those wires the end of the long lever is drawn towards the wall; the catch upon the rod W is released, and the wheel stopped as before.

Action of the Governor.

Although, in the foregoing detailed account of the governor, we have described its mode of action in cases of emergency, its ordinary operation is still unexplained, and it is of so peculiar a nature as to render some brief account necessary.

The normal velocity at which the governor spindle i, is driven, is about 40 revolutions per minute, and at this rate the balls G, G are held suspended by the centrifugal force, in such a position that the small eccentric n, revolves between the two catches o, o, without coming into contact with either, and the shaft q, remains at rest. If, however, the speed exceeds this rate, the balls diverge in consequence of the increased centrifugal force; the slide m, is elevated, and the eccentric strikes the upper catch o. The levers H, I and p, are thus set in motion; the clutch-box K is thrown into geer with the wheel f, and so long as it continues so engaged, the shaft q receives its motion, which being transmitted through the worm-wheel Q to the sluice-geering, partially closes the sluice, and thereby diminishes the velocity of the wheel. In consequence of this diminution of velocity, the governor balls converge, the slide m, falls, and the eccentric striking against the lower catch o, the clutch-box K is disengaged, and the motion of the shaft q, ceases. Should the velocity of the wheel, on the other hand, become too low, the convergence of the balls brings the eccentric into contact with the lower catch, which throws the clutch-box into geer with the wheel g; a motion in the contrary direction to the former is imparted to the shaft q, and the quantity of water admitted at the sluice is increased.

By this mode of action, it is obvious that no strain whatever is thrown upon the governor, beyond the slight amount of power necessary to throw the clutch-box into and out of geer with the small bevil wheels; and thus the delicacy of operation so requisite in all regulators of prime movers is ensured.

Two clocks, (not shown in the drawings), are affixed to the wall of the wheel-house, immediately over the governor, for the convenience of estimating and registering the work performed by the wheel. One of these is a common pendulum clock, and the other is a combination of clock-work set in motion by the small shaft z, and so arranged that when the wheel performs its prescribed number of revolutions they should both coincide. The amount of surplus or deficiency may be read off at the end of each day by comparing the dial-plates of each.

Literal References to the Wheel.

A, A, A, the shrouding.
a, a, a, joints of the shrouding.
B, B, B, the buckets.
c, c, c, flanges on the shrouding, to which the buckets are riveted.
C, C, C, sole of the buckets.
d, d, d, flange on the shrouding, to which the sole is riveted.
e, e, e, angle-irons by which the buckets are riveted to the sole.
f, f, f, the stay-bolts between the buckets.
D, D, D, the arms of the wheel.
g, g, g, bosses cast on the shrouding for the insertion of the arms.
E, E, E, the diagonal braces.
h, h, h, bosses cast on the shrouding for the diagonal braces.
F, F, the centres or naves.
i, i, i, sockets cast on the centres for the insertion of the arms.
j, j, j, sockets cast on the centres for the diagonal braces.
G, the cast-iron shaft or axis.
k, k, the keys for fixing the shaft to the centre.
H, H, the main-centre plummer-blocks.
l, l, grease-boxes for lubricating the shaft.
m, m, eccentrics for raising the wheel.
n, n, the sole-plate of the plummer-blocks.
o, o, o, the holding-down bolts.
I, the great internal spur-wheel.
J, the pinion working into the great internal spur-wheel.
K, the second motion spur-wheel.
L, the pinion on the main shaft into the mill.
M, M, the cistern or trews for conducting the water to the wheel.
p, p, the sluice-face of cast-iron.
q, q, the sluice-cover.
r, r, racks for working the sluice.
s, t, u, v, w, the wheels and pinions of the sluice geering.
x, shaft from the sluice geering to N, the governor.
y, z, shaft and geering for driving the governor.

Literal References to the Governor.

A, the main bracket to which the governor is fixed.
B, B, the standards forming the framing of the governor.
a, *a*, bearings in the standards for the cross shaft.
b, *c*, show the mode of junction of the standards B, B.
C, C, cross-stays between the standards for supporting
D, the piece in which the lower end of the governor spindle revolves.
d, *d*, the upright shaft which drives the governor.
E, the upper bearing of the shaft *d*.
e, *f*, *g*, *h*, a system of meter wheels driving the governor.
i, *i*, the governor spindle.
J, a brass socket, to which are jointed
k, *k*, the legs of the governor balls.
F, a cross arm forming a guide for the legs *k*, *k*.
G, G, the governor balls.
l, *l*, links connecting the legs *k*, *k*, with
m, *n*, a brass slide and eccentric.
o, *o*, catches against which the eccentric *n*, strikes.
H, I, J, *p*, levers and axis for conveying the action of the eccentric to
K, a clutch-box sliding upon the cross shaft.
q, *q*, the cross or horizontal shaft of the governor.
L, a sliding coupling-box for throwing the governor into and out of geer with the sluice.
M, a bracket forming the centre of motion of
r, the lever for working the coupling-box L.
s, a small rod and handle for working the lever *r*.
N, the endless screw or worm.
O, a bracket for supporting the outer bearing of the shaft *q*.
P, hand-wheel for starting and stopping the water-wheel.
Q, the worm-wheel on the shaft leading to the sluice geering.
R, the carriage in which the bush of the worm-wheel shaft works.
S, a lever-handle for raising or depressing the end of the worm-wheel shaft by hand.
t, *u*, *v*, *w*, *y*, *z*, a system of wheels and shafts for closing the sluice quickly.
T, U, brackets on which the bearings of the shafts *v*, and *z*, are supported.
V, a cast-iron drum revolving under the worm-wheel.
W, an upright rod jointed to
X, a lever for raising or depressing the end of the worm-wheel shaft by the action of the machine itself.
Y, a long lever acting upon the upright rod W, and connected by wires to the various flats of the mill.

PLATES XCII., XCIII.

BREAST WATER-WHEEL. BY WILLIAM FAIRBAIRN, MANCHESTER.

The researches of scientific mechanics within the present century have effected few more marked improvements in the construction of water-wheels than in demonstrating that the stream of water should, in all cases, be led on to the wheel at a point in its circumference considerably *below* the summit, and not *over* it. By this arrangement, a wheel of a much greater diameter for a given fall than could otherwise be employed may be erected without any loss, or rather with a positive increase, of power from this cause. The distinction between over-shot and breastwheels is thus to a certain extent abolished, and the use of these terms is rendered, in some measure, obsolete. Still, however, there remains one great distinctive peculiarity to warrant their use, and accordingly we have given an example of what may, with the strictest propriety, be denominated a *breast-wheel*.

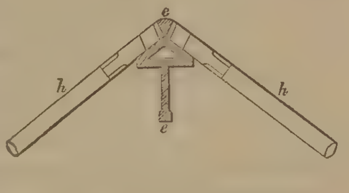
The Plates now under consideration give a general delineation of the form and dimensions of a wheel erected in the year 1842, to drive the machinery of a flax mill, the property of Thomas Ainsworth, Esq. of Cleator, near Whitehaven, in Cumberland. Like the water-wheel which formed the subject of the Plates immediately preceding, it is remarkable for its great dimensions; and as the former is distinguished above all others of its class for its great diameter, the present wheel is remarkable for its unusual breadth. It was constructed by Mr. Fairbairn of Manchester, who has devoted much of his attention to the subject of water-wheels, and who has introduced into it his most recent improvements in the ventilation of the buckets.

Fig. 1 represents a front elevation of the water-wheel, with a portion of the side shrouding and segments removed, in order to show a section of the buckets, and the arrangements for the ventilation. This view also exhibits a section of the *breast* against which the wheel works, together with the form and arrangement of the sluice and its geering. In it we have also introduced a full sectional elevation of the wheel-house, principally for the purpose of showing its very handsome and substantial iron roof.

Fig. 2 is an end elevation of the wheel, one half being in section to show the form of the shaft, the arrangement of the arms and diagonal braces, and the mode in which the power is transmitted.

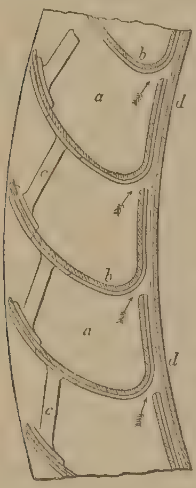
General Description.

This wheel is about 20 feet in diameter, by about 22 feet in breadth. With the exception of the ventilating apparatus and the diagonal braces, its constructive details are so similar to those of the wheel last described as to require no separate notice. Between the buckets *b*, *b*, *b*, cast-iron stays *c*, *c*, *c*, are employed instead of the stay-bolts used for the same purpose in the Greenock wheel. On account of the great breadth of the wheel, it was necessary to give further strength and rigidity to it than could be imparted by the side shroudings *a*, *a*. Accordingly, a cast-iron ring *e*, *e*, is fixed to the sole-plate *d*, *d*, at the middle of its breadth, and this ring is braced to each of the centres *g*, *g*, by eight wrought-iron rods *h*, *h*, *h*, fixed by cotters to the sockets *i*, *i*, cast on the inner surfaces of the centres, and by T-heads to the ring *e*, the form of which and the manner in which the ends of the braces are attached, will be clearly understood by reference to the annexed wood-cut. The arms *f*, *f*, are jointed to the shrouding and centre precisely in the manner described in treating the previous subject.



The distinguishing feature which we have alluded to in the commencement of this description, consists in the wheel working against a circular *breast*, or facing of solid masonwork *r*, *r*, concentric with it, and fitted close to the outer circumference of the shrouding and ends of the buckets. The cast-iron face of the cistern *s*, *s*, leading the water to the wheel, is likewise fitted close to the wheel, and forms in fact a continuation of the circle of the breast. Thus the entire loaded portion of the wheel is enclosed on all sides, and in forming the buckets it is not necessary, as in the case of the over-shot wheel, to contract the mouth for the purpose of retaining the water, since this object is attained by their close contact with the breast-work itself.

All water-wheels with close buckets are more or less subject to an inconvenience which, unless an effectual remedy be applied, would greatly impair their useful effect. In cases where the mouth of the buckets is so contracted that the water is discharged upon them in a greater section than the opening between them, it is obvious that, when the empty buckets arrive at the sluice, the air contained in them will be suddenly condensed by the weight of the water, and again re-acting by its elastic force, will throw it back upon the orifice of the cistern, and thus allow the buckets to pass without being sufficiently filled. To remedy this defect which, as we before remarked, presents itself in all wheels of this class, Mr. Fairbairn has applied various systems of ventilation, and in the wheel now under consideration, has introduced his most improved method. This consists in forming the buckets *b*, *b*, and sole-plate *a*, *a*, so as to leave a small space between them all round the wheel, while a communication is opened between this space and the interior of each bucket. The annexed figure illustrates the nature of these arrangements. By referring to it, it will be seen that a free passage, in the direction of the arrows, is provided for the escape of the air during the process of filling, and that consequently the water will fall unimpeded into the buckets. A small quantity of water will sometimes escape with the air above the lip of the outlets; but that is of little consequence, as it is again returned to the buckets as the wheel moves round.



In the transmission of the power of this wheel, two great internal spur-wheels *l*, and *p*, are employed, the former of which is shown in section through the flanges of the segments. The great breadth of the wheel renders this arrangement necessary for the sake of balancing the strain; this object is further ensured by placing the pinions *m*, and *q*, which transmit the power of the wheel, the one above and the other below its centre.

The water is laid on by a circular cast-iron shuttle *u*, which is fitted water-tight to the face *t*, *t*, prepared to receive it, and which forms the end of the cistern *s*, *s*, by which the water is conveyed to the wheel. To this shuttle racks and pinions are attached, as described on a former occasion, and by a train of connecting shafts and wheels, the governor regulates the supply of water which flows over the shuttle into the buckets. The governor of this wheel is, in all essential particulars, identical with that formerly described.

Literal References.

a, *a*, *a*, the shrouding.
b, *b*, *b*, the buckets.
c, *c*, *c*, the stays between the buckets.
d, *d*, *d*, the sole-plate.
e, *e*, *e*, the middle ring.
f, *f*, *f*, the arms of the wheel.
g, *g*, *g*, the sockets on the centre for the insertion of the arms.
h, *h*, *h*, the diagonal braces.
i, *i*, *i*, the sockets on the centre for the diagonal braces.
j, *j*, the main centre shaft.
k, *k*, the plummer-blocks in which it revolves.
l, *p*, the two great internal wheels.
m, *n*, *o*, *q*, the geering by which the power is transmitted.
r, *r*, the breast-work.
s, *s*, the cistern in which the mill-race terminates.
t, *t*, the sluice-face.
u, the shuttle or sluice-cover.
v, *v*, racks for working the shuttle.
w, *x*, the sluice-geering.
y, the tunnel of the tail-race.

PLATES XCIV., XCV.

PONCELET'S UNDER-SHOT WATER-WHEEL. CONSTRUCTED BY M. M. BORNET AND BEUVAU.

In those localities where the supply of water is large and the fall comparatively low, the under-shot water-wheel offers an obvious, and perhaps the best, as well as cheapest medium of transferring the power of the current, and of rendering it available for manufacturing purposes. Its frequent application, arising from these circumstances, and the low ratio of effect transmitted by the common radial-floated wheel, induced M. Poncelet, about 20 years ago (1825), to examine scientifically the conditions upon which the efficiency of this species of wheel depended, with a view to ascertain whether it might not be rendered more economical in its action. The investigation, which embraced a very extensive series of experiments, led to the discovery that the floats, instead of presenting a plane radial surface to the impulse of the water, ought to consist of circular arcs of assignable radii of curvature, and ought, moreover, to be disposed upon the circumference of the wheel, at an angle defined by the condition that the greatest possible amount of the *vis viva*, or inherent energy of the moving current, shall be rendered available upon the wheel at the moment of impulse. The necessity of departing from the received practice was fully demonstrated by mathematical considerations, and confirmed by the successful substitution of a wheel constructed on the new principle developed and recommended by M. Poncelet, in place of one of the old form, in a situation and under circumstances which were deemed well adapted to test the comparative efficiency of both. Many others have since been constructed in accordance with the same rules, and with equal success. Among the number is the example represented in these Plates, which was erected upwards of seven years ago, at the French Government Iron Works of Guèrigny in Nièvre, for the purpose of driving the rolling mills used in the manufacture of cable iron. Its diameter is 22 feet, and the weight 13½ tons. It makes 8·81 revolutions per minute; its velocity is, therefore, 10 feet per second at the circumference, which, compared with the velocity due to the head of water under which it acts, is as 3 to 5. The nominal power is 30 horse, and the cost of construction and erection was L.814.

Plate XCIV.—The two drawings in this Plate show a side elevation and plan of the wheel, sluice, and framing, partly in section.

Plate XCV. contains the constructive details of the wheel, which will be enumerated and described in the sequel.

General Description.

The wheel is constructed entirely of iron. It is carried on a strong framing of wooden beams, supported on one side by the wall of the building into which the power is transmitted by the wrought-iron axle E of the wheel. On this axle the three cast-iron centres D, D, D are keyed; and each of these carry eight radial arms C, C, C of malleable iron. The arms are braced together by the diagonal and transverse tie-rods *c*, *c*, and *d*, *d*, at the middle of their length, and have their outer extremities flattened, to allow the shrouding plates to be riveted upon them flush with the lateral surface. The wheel has 48 double buckets B, B, formed of sheet-iron 3-16ths of an inch thick. Each compartment is inclosed by a single plate stretching between the two contiguous shrouds, to which it is attached by bolts at each end, by means of cast-iron flanges *b*, *b*, riveted to the shroud-plates; these flanges having the proper form of the buckets, and serving to support their extremities. The shroudings are formed of strong iron plates, cut into segments, which are joined together by laps and rivets. The middle shrouding is introduced, in order to give the necessary support and rigidity to the bucket-plates without the use of intermediate stays, which ought, as far as possible, to be avoided in works of sheet-iron exposed to water, as they speedily corrode and destroy the iron round the palms by which they are attached.

The extreme diameter of the wheel, as before stated, is 22 feet. This dimension is regulated by the height of the head of water which can be obtained in the situation, and other local circumstances which may influence the relative position of the works to be impelled. The depth of the shroudings depends likewise upon the available fall; but this dimension also admits of modification within certain limits. According to the rules laid down by M. Poncelet, a good proportion for falls of 6½ feet and upwards, is a fourth of the whole height; but for lower falls, the depth may, without disadvantage, be augmented to a third, and even a half of the entire head. The breadth of the wheel depends, of course, upon the volume of water to be expended.

The reservoir and floor of the water-course H, H are constructed entirely of timber, and rest upon a strong framing of the same material, carefully constructed, at certain points, according to determinate mathematical rules. The area of the aperture by which the water is admitted upon the wheel is regulated by an inclined sluice-plate J, worked by means of the winch K. This winch acts by intermediate geering upon the rack L, which is attached to the wooden frame M, M, bolted at its lower end to the sluice-plate, and supported at the upper end by guide-rollers. The sluice-plate J is further guided by sliding in grooves formed in the side-beams N, N of the apron. To give additional strength to the planking of which the apron mainly consists, and to render it capable of withstanding such occasional pressures of water as may result from excessive augmentations of the stream by floods, a cross binder O is fixed on the back of the planking at the middle of the width, and abuts securely against the strong transverse beams P, P.

Water, in passing through an aperture, always suffers contraction of its volume in proportion to the area of the orifice through which it flows. To compensate for the contraction due to the efflux of the fluid by the sluice, or rather to render it as small as possible, the mill-race is made of greater width than the breadth of the wheel, and is contracted, in its approach to the sluice-opening, conformably to the known mathematical law by which the discharge is the greatest possible for a given area of orifice. The curvilineal outline of this part is indicated by the dotted lines *x*, *y*, in fig. 2.

In order that the water may impinge upon the buckets of the wheel with the least possible loss of impulsive force, the bottom of the course, where it approaches the point of action, is made with a declination of from 3 to 4 degrees from the horizontal direction. The line of direction *s r*, of that portion of the water flowing through the sluice-opening J, ought therefore to make with the tangent line *r t*, to the circle of the wheel at that point, an angle of 24 to 25 degrees. The surface *u v*, of the water-run is a tangent drawn to the small circular arc *u w*, which approaches close upon the circumference of the wheel at that part where it passes the vertical plane, and which should be at least as long as the interval between two contiguous buckets. At the point *w*, the arc terminates abruptly throughout the entire breadth of the water-run, allowing a descent for the expended water equal to half the depth of the shrouding.

Plate XCV.—This Plate contains constructive details of the wheel, and of the apparatus by which the bucket-plates are bent to the required form.

Figs. 1 show the mode of attaching the shrouding upon the arms, the extremities of which are flattened and rebated to the depth of the shrouding plates. By this provision, the shroudings are supported independently of the rivets, and a more finished appearance is given to the work than if the plates projected beyond the plane of the arms.

Figs. 2 show the mode of fixing the bucket-plates to the shroudings. The cast-iron runners *b*, *b*, are formed with suitable projections through which the holding bolts of the bucket-plates pass. The buckets are retained in their places by these bolts, and are also sustained, along the whole depth of the shrouds, by the projecting concave ledges of the segments upon which they rest. The form of these segments is further shown by figs. 3 and 4, the latter being a partial sectional plan, taken at one of the points through which the attaching bolts pass.

Fig. 5 shows the manner of staying the shroudings. The rod *e*, *e*, is in length equal to the width of the wheel, and passes through the three shrouds parallel to the axis; and these are retained at their proper distances apart, by a nut on either side of the plate. The screwed parts of the rod are increased in diameter, in order to allow the nuts for the central partition of the shrouding to be passed along the rod. Fig. 6 is one of a similar set of stay-rods for giving rigidity to the arms of the wheel.

Fig. 7. represents the eye of the wheel, and the mode of fixing the arms to it. The extremities of the arms are received into radial recesses cast in the eye, of proper depth to allow them to sink flush with the general surface. They are fixed by two bolts and two keys, one on each side, their extremities abutting upon the bottoms of the recesses. This arrangement is also partially shown by fig. 8, which is a cross section on the dotted line in fig. 7; and more fully by the general section (fig. 9), through the axis of the eye.

Figs. 10 and 11 represent the apparatus used to bend the bucket-plates to the required curve. This curve being the segment of a circle, (the radius of which is determined by a rule which will be given below), the operation of bending the buckets may be greatly facilitated by a simple mechanical contrivance. This consists of a sector of wood, which is prepared and firmly attached to a strong timber framing R, R, at its two extremities. The height of the sector ought to be at least equal to the surface depth of the bucket, and its length equal to that of the plate to be bent. It will, however, be found advantageous to make its radius of curvature 1-5th less than that of the bucket, in order that the elasticity of the plate may tend, in some measure, to retain it in close contact with the flanges upon which it is to be attached; this provision further allows of a certain amount of deviation from the mould curve, which always takes place in consequence of the elasticity of the plate being only partially overcome in the process of bending.

A strong wooden roller S, carried in a movable frame of wrought iron T, T, T, working on an axis U, is applied to the sector, leaving only sufficient space between to receive the plate to be bent. The axis U being, moreover, in the centre from which the curve of the sector is described, when the roller S is moved vertically by the working handle *h*, attached to the frame, its inner surface will describe an arc *f g*, similar to, and concentric with, that of the sector; and therefore if a plate be placed between these surfaces, with one of its edges fixed to the lower extremity of the arc, it will be bent into that form, and if quite inelastic would retain the exact curvature of the mould. This is the operation performed with the mechanism under notice; but the partial elasticity of the plates requires that it be several times repeated before the requisite degree of curvature is permanently impressed upon them.

The plate under the process of bending is attached at its lower edge by an iron bar V, which is tightly driven up by two wedges *i*, *i*, passed behind it in the slots of the iron pieces *j*, *j*, which are firmly attached to the ends of the mould Q, Q. This secures the edge of the plate, and prevents it from springing back, while the roller S is made to pass over its surface in a vertical direction, by manual power applied to the handle *h*.

Rule for determining the Curvature of the Buckets.—Plate XCV., Fig. 2. The form of the buckets, as before intimated, ought to be that of a circular arc, and may be thus traced:—From a point *k*, in the outer circumference of the wheel, draw the line *k p*, making an angle of 25 degrees with the radius *k l*, drawn from the centre of the wheel to the point *k*, in the circumference. Then with a radius of about 1-6th of the depth of the shrouding less than that of the inner circle of the shrouding, describe from the centre another circle *m n*, cutting the line *k p*, in *p*; this point *p*, will be the centre from which the curve *q k*, of the bucket ought to be drawn.

Literal References.

A, A, the shrouding-plates.
a, *a*, the half-lap joints of the shrouding-plates.
B, B, the buckets.
b, *b*, cast-iron flanges, to which the buckets are attached.
C, C, the arms of the wheel.
c, *c*, stay-rods between the arms.
d, *d*, *e*, *e*, transverse stay-rods for strengthening the wheel.
D, D, the centres or naves of the wheel.
E, E, the shaft or axle.
F, F, the plummer-blocks.
G, G, the frame-work of timber on which the plummer-blocks rest.
H, H, the planking of the mill-race and tail-race.
I, a beam in which is formed an abrupt descent for the liberation of the water.
J, the sluice-plate or cover.
K, winch for working the sluice.
L, a rack conveying the motion of the winch to
M, M, a timber frame working in guide-rollers, and bolted to the sluice-plate J.
N, N, the side beams, grooved to receive the sluice-plate.
O, a beam for strengthening the planking of the apron.
P, P, the transverse beams of the apron.
Q, R, S, T, U, V, parts composing the machine for bending the buckets.
f, *g*, *h*, *i*, *j*, smaller parts of the same machine.

PLATES XCVI.—XCVII.

WATER-SLUICES.

In connection with the subject of water-wheels and water-mills, which we have treated so fully and illustrated in numerous Plates immediately preceding the present, we have also introduced drawings and descriptions of several varieties of that important adjunct to all hydraulic machines, the sluice; and these we have represented sufficiently in detail to give a clear idea of the means employed for feeding the mill with water, and for regulating the supply. This is, however, a subject of such importance that we feel convinced that a few more examples, adapted to circumstances different from those which have hitherto come under our consideration, will not be unacceptable to our readers, and accordingly we have given, in the Plates now before us, several other varieties of the water-sluice. We have also added, in figs. 15–17, a representation of the very ingenious mechanism invented and extensively employed by the late Mr. Thom of Rothsay, for maintaining the surface of water in a mill-dam or water-course uniformly at a given level.

Plate XCVI. is occupied with a design for a water-sluice of considerable dimensions, suitable for the *head* of a mill-race, or that point where it emerges from the mill-dam. It is also, with slight modifications, applicable to lock-gates for canals, docks, &c.

Fig. 1 is an elevation of the back of the sluice, looking in the direction of the current. One of the boxes in which the sluice-geering is enclosed is shown in section, that the arrangement of the geering may be exhibited, while the other is shown in an external elevation.

Fig. 2 is a vertical section taken through the centre of one of the compartments of the sluice.

Figs. 3 and 4 are sectional plans, the first taken near the centre of the framing of the sluice, and the other affording a view of the arrangement of the sluice-geering.

Figs. 5, 6, and 7, are detailed views, on an enlarged scale, of some of the more minute parts of the sluice-geering.

The framing A, A of the sluice consists of two separate castings, each of which is in the form of an oblong cast-iron plate, strengthened by feathers so arranged that those on one side may be used as flanges for bolting the two together (see fig. 3); the feathers on the exterior and lower edges of the plates forming a species of box, fitted accurately within the stone work of which the sides of the sluice-bed are composed, while the plates themselves project beyond, so as to afford the means of bolting them to the face of the masonry. The feathers of the upper surface of the sluice-framing are expanded into a species of flat table or sole-plate for the attachment of the bearings of the various parts of the sluice-geering.

The lower portions of the sluice framings A, A, are formed with rectangular apertures of such dimensions that the sum of their areas shall be equal to that of the greatest body of water that may at any time be required to pass through the sluice. The edges of these apertures are provided with fitting-strips accurately planed, and these are extended above the upper edges to a height at least equal to the depth of the apertures. The cast-iron sluice-gates B, B are fitted water-tight upon these faces, and slide vertically between projecting guides cast upon the sluice framing. They are strengthened by feathers, and in the middle of each is formed a socket for the reception of a strong malleable iron rod *a*, by which the motion of the sluice-geering is conveyed to the gate. The rods *a*, *a*, are secured in their places by gibs and cotters, and, to counterpoise the sluice-gates so as to facilitate the working of them, the two cast-iron weights C, C, are attached by chains passing over the pulleys *b*, *b*, to the lower ends of the rods *a*, *a*.

These rods may be made of any length to suit the circumstances in which the sluice may be required to act; in the case of a lock-gate they will, of course, be very much longer than in the drawings now before us. To the upper extremity of each is bolted a cast-iron rack *c*, geering into a pinion *d*, fast upon an axis, (fig. 5), having its bearings in the cast-iron cheeks D, D, which are bolted to the flanges of the sluice framing. The rack and rod to which it is attached are guided in their vertical motion, and the former retained in geer with the pinion *d*, by means of a friction pulley *e*, whose axis is formed in a small stud-bracket, (figs. 5 and 6), in which are three elongated bolt-holes for the purpose of adjusting the rack into its proper position in relation to the pinion. On the same axis with the pinion *d*, is fixed the worm-wheel *f*, geering with an endless screw *g*, which is fast upon the shaft *h*, working in the small plummer-blocks E, E, and prolonged on the one hand to the outside of the sluice framing, where it is furnished with a hand-wheel F, and on the other to the interior of the remaining box D, which contains geering in all respects similar to that which we have described.

From the above description, and from inspection of the drawings, the mode of working the sluice will be very obvious. The two endless screws are set in motion by causing the hand-wheel F to revolve. These impress a slow and powerful motion upon the worm-wheels *f*, *f*, which, by means of the pinions *d*, *d*, is communicated to the rack *c*, *c*, and rods *a*, *a*; and thus both the sluice-gates B, B, are simultaneously raised or depressed; with this very obvious advantage over the single sluice-gate, that there is much greater facility in keeping them water-tight, and that much lighter and more manageable geering may be employed in working them.

The end plates *i*, *i*, of the boxes D, D, in which the geering is enclosed, should be made of sheet-iron, hinged below and secured above by small pins passing through studs *j*, *j*, riveted into the cross bars which serve to bind together the two cast-iron cheeks of the boxes. Thus ready access is afforded for the examination and repair of the geering. The covers of the boxes may also be made of sheet-iron inserted into grooves cast on the insides of the cheeks, as shown in figs. 1, 2 and 4.

A plank G, G, placed at a convenient height for working the sluice, is usually thrown across the mill-stream.

Plate XCVII.—Figs. 8, 9, 10, and 11. These figures represent external and sectional views of two species of water-sluices, adapted for regulating the supply of water conveyed in pipes, when, on account of their size, cocks could not be employed. For water-mills or turbines they are especially suitable. Fig. 8 is an external elevation, and fig. 9 a longitudinal section of a sluice suited to circumstances similar to those in which that last described is placed, with this difference, that instead of an open mill-stream, a close pipe is, in the present instance, used for conveying the water. Figs. 10 and 11 show corresponding views of a form adapted for the *middle* of a pipe where there is a considerable pressure of water acting upon it; but these two forms are so similar in their general construction, that we shall describe both by reference to the same letters.

H is the face against which the sluice-cover I works; in the form which comes first in order, it consists merely of a broad and elongated flange cast on the end of the pipe; but in figs. 10 and 11, it is formed on one side of an oblong box cast in the middle of a short length of pipe, and close on all sides, except the top, where it is surmounted by a water-tight cover on which the upright standards J, J, are cast. The sluice-rod *l*, *l*, passes through a stuffing-box in this cover; and its upper portion is formed into a screw of sufficient length to admit of the full travel of the sluice-plate I. The screw works into a cylindrical brass nut *m*, fitted to revolve in bearings which may be either cast or bolted to the upright standard J. The nut *m*, has a strong collar bearing against the under side of the plummer-block, and is retained in its place by being keyed into the eye of the hand-wheel F, which works against the upper surface. By this arrangement it is obvious that on turning the hand-wheel in either direction the rod *l*, and sluice-plate I, to which it is attached by cotters, will be raised or depressed. The holes on the back of the sluice-plate through which the rod passes are made of an oblong form, in order to admit of its being kept constantly pressed against the working face without straining the rod.

Figs. 12, 13, and 14 are views of another kind of water-valve, which may be employed with advantage in circumstances similar to those to which the form showed in figs. 10 and 11 is applicable, as for water-mills or turbines. The present form is peculiarly adapted to pipes of very great dimensions, and may be worked with equal convenience in whatever position the pipe may be placed. At the point where the sluice is required to be situated, the pipe is made to pass from a round into an oblong form; and the valve-box K is bolted water-tight to it. Within the valve-box is cast a strong diaphragm or face *h*, perforated with numerous oblong holes. This is accurately dressed and fitted water-tight to the sluice-plate or valve *o*, in which are formed apertures of corresponding dimensions. The width of these apertures and of the portions of the valve and sluice-faces contained between them is so arranged, that when the valve is depressed to its lowest point, the ports in the sluice-face shall be entirely covered, and slightly overlapped by the valve-face. When, on the other hand, the valve is raised to its highest point, the ports in both faces will coincide, and a free passage be left for the water. It is almost unnecessary to remark, that the area of the ports should be at least equal to that of the main body of the pipe to which this species of sluice is applied.

When the dimensions of a sluice of this kind are great, it may be advantageously worked in the manner indicated in our drawings. Here two valve or sluice-rods *p*, *p*, are employed, which are simultaneously acted upon by geering similar to that used in the form which first came under notice. These rods are screwed throughout their entire length, the lower portions to admit of their being secured in their places by nuts binding them to the feathers cast on the back of the valve, by which means great strength and rigidity is imparted to it; while their upper extremities are fitted into the cylindrical brass nuts *q*, *q*, which work in water-tight stuffing-boxes, and are provided with broad collars to prevent their rising out of their places. To the upper extremities of the nuts *q*, *q*, the worm-wheels *r*, *r*, are keyed, and motion is imparted to them by the endless screws *s*, *s*, fixed upon the axis of the hand-wheel F.

Figs. 15, 16, and 17. These figures represent views of a highly ingenious and useful contrivance, which may be termed a *self-acting sluice*, or *mill-stream governor*. Its object is to prevent the water within a mill-race, or any collection of water whatever, whether at rest or in motion, from rising above or falling below an assigned level.

Closely contiguous to the cistern or embankment L, in which the water whose level it is required to regulate is contained or conveyed, is placed an oblong box M, which may be made either of wood or iron. This box has a free communication, by means of the pipe *t*, with the water contained in L, and within it is enclosed a bent pipe *u*, bolted to one side of the box, and terminating in a funnel, the lip of which is situated exactly at the level at which the water is to be maintained. On the outside of the box M, and in continuation of the pipe *u*, is bolted another pipe *v*, cast upon a large hollow cylinder N, close at the bottom, near which another small pipe *w*, branches off, conveying the water which may be contained in the cylinder to a lower level than that of the bottom of the cistern L. Within the cylinder N is placed another hollow water-tight cylinder O, moving freely in it, and attached by means of a chain passing over a pulley *x*, to the lower edge of a strong cast-iron flap-valve P, fitted against the opposite side of the cistern L. The cylinder O is of sufficient weight, when unsupported by water, to keep the valve P firmly closed against the pressure of the water within the cistern L. The grating *y*, serves to protect the valve from being obstructed in its action by pieces of wood or other bulky substances which may be contained in the water.

When the water in the cistern L rises above its prescribed level, it flows over the lip of the funnel *u*, and passes into the cylinder N, and this diminishes the weight of the cylinder O so much that the pressure of water upon the front of the valve P throws it open and admits of a copious escape. When the water subsides so as no longer to flow into the pipe *u*, the cylinder N is emptied by the pipe *w*, and the weight O being unsupported, the sluice P is closed, and the water retained within the cistern. By properly proportioning the diameters of the pipes *t*, *u*, *v*, and *w*, to each other, and the weight of the cylinder O in relation to the column of water pressing upon the sluice P, the greatest accuracy and precision in the regulation of the water may be attained by means of this simple apparatus.

Literal References.

Plate XCVI.

A, A, the sluice-framing.
B, B, the sluice-gates or covers.
a, *a*, the wrought-iron sluice-rods.
b, *b*, pulleys, over which pass the chains for suspending
C, C, the counter-balance weights
D, D, boxes enclosing the sluice-geering.
c, *c*, racks attached to the upper ends of the rods *a*, *a*.
d, *d*, pinions working in the racks *c*, *c*.
e, *e*, friction pulleys for guiding the rods *a*, *a*.
f, *f*, worm-wheels working into
g, *g*, endless screws fixed upon the shaft *h*, working in the bearings E, E, and moved by the hand-wheel F.
i, *j*, doors and fastenings of the boxes D, D.
G, a plank placed across the mill-stream for the convenience of working the sluice.

Plate XCVII.

H, H, the casings or boxes of the pipe-sluices.
I, I, the sluice plates or valves.
J, J, stuffing-box and guides for the bearings of the sluice-rod.
k, *k*, guides for the vertical motion of the sluice-plate.
l, *l*, the sluice-rods, and *m*, *m*, the nuts in which they work.
K, oblong cast-iron box for great pipe-sluice.
n, the sluice-face or diaphragm.
o, the valve or cover.
p, *p*, the valve-rods.
q, *q*. cylindrical brass nuts, through which the valve-rods work.
r, *r*, *s*, *s*, worm-wheels and worms for working the sluice.
L, a cistern, mill-stream or other vessel containing water.
t, a short pipe leading to the interior of
M, a box communicating freely with the cistern L.
u, *v*, short pipes leading to the interior of
N, a cylindrical cast-iron vessel.
O, a hollow cylinder suspended in water, and attached by a chain passing over a pulley *x*, to the lower edge of the flap-valve P.
y, a grating to protect the valve.

PLATES XCVIII.—CI.

IMPROVED CORN MILL. BY WILLIAM FAIRBAIRN, MANCHESTER.

The operation of reducing the fruits of the harvest into a state fit for becoming the food of man, is naturally the earliest in point of date, as well as the first in point of importance, to which the aids of mechanical science have been applied; and although the machinery required for effecting it is not, perhaps, of such a nature as to admit of that indefinite improvement of which many of the processes in the mechanical arts are susceptible, it has, nevertheless, from time to time, undergone important ameliorations.

In the process of grinding corn, modern practice differs but little, in its main features, from that which has been in use for many centuries: the corn being caused to pass between two horizontal stones, placed nearly in contact with each other, and furrowed by grooves on their contiguous surfaces; the lower stone being immovable, while the upper revolves upon a spindle, and has a hole in its centre through which the corn is admitted. Various modifications of this method have been proposed, and, to a limited extent, adopted; such as the substitution of steel cutters for furrows in the stones, of annular for cylindrical, and, more recently, of eccentric for concentric stones. None of these methods, however, appears to possess such advantages as to justify its adoption in preference to the established practice; which, simple as we have announced it to be, has yet, in its minor details, partaken so largely of the mechanical improvements of modern times, as to have changed the character of the corn-mill from a rude and unwieldy combination of timber, stone, and iron, into a highly elegant and efficient piece of machinery. We have, accordingly, in laying down the plan of this work, included this important department of the Millwright's art, and now proceed to lay before our readers a series of Plates partly illustrative of the general arrangement, and partly furnishing minute and accurate details, of that form of corn-grinding machinery which is most extensively in use at the present day.

Among the mechanicians who have contributed to the improvement of that description of machinery now under consideration, Mr. Fairbairn of Manchester occupies a most distinguished position, and accordingly we find that he is most extensively employed in the erection of corn-mills throughout all parts of Great Britain and the Continent. The example which we have chosen for detailed illustration is a small mill of three pairs of stones, erected by him for His Excellency the Seraskier Halil Pacha, of Constantinople. It is interesting as exhibiting, within a small compass, an epitome of all the processes carried on in larger establishments, and thereby enabling us to show the mechanism by which these processes are effected, on a larger scale than could otherwise be possible. It possesses, moreover, an interest independent of this, and which we feel confident will be appreciated by our readers: the circumstance, namely, of its being contained within an iron house, the construction of which our drawings enable us to exhibit with tolerable distinctness.

Enumeration of the Plates.

Plates XCVIII., XCIX.—Fig. 1 is a sectional elevation of the mill, the line of section being taken in a longitudinal direction, and exhibiting the position of the stones, the engine and driving geering, and of such portions of the subordinate apparatus as are visible on the side of the mill which is exposed to view.

Fig. 2 is a sectional plan corresponding to the above, and taken on a horizontal line passing through the lower story of the mill.

Plates C., CI.—Fig. 3 is a transverse section of the entire mill, in which are shown the garners for undressed and dressed wheat, the mechanism by which it is cleaned and conveyed from the former into the latter, the sack-tackle, &c.

Fig. 4 is a sectional plan corresponding with the plan fig. 2, the line of section being taken through the second story of the mill.

General Description.

The house in which this mill is contained consists of an assemblage of plates of sheet-iron A, A, A, of a suitable thickness, consolidated and bound together by the cast-iron columns B, B, B, and by the strong cast-iron girders C, C, C, situated at such a height as to oppose and neutralize the strain of the principal working parts. It is surmounted by an arched roof D, D, formed of plates of corrugated sheet-iron. A wall of masonry E, E, is erected in the interior, for the purpose of affording a foundation for the bearings of the heavier geering of the mill. The motive power is supplied by a high-pressure steam engine F, of 12 horse power, of that peculiarly compact and elegant description introduced by Mr. Fairbairn, the distinguishing feature of which consists in its principal working parts being wholly inclosed within a large cast-iron column. By this arrangement great firmness and stability is imparted to the engine, while the space which it occupies is reduced to the smallest possible dimensions. The boilers G, G, are situated in an adjoining part of the house, and their flues H, H, are formed, in the usual manner, of brickwork, abutting on the one hand against the wall E, and on the other against the side of the house itself. Thus the engine and boilers occupy nearly the entire half of the lower story of the mill. The whole erection is strengthened and bound together by the cast-iron beams I, I, I, which pass transversely through the interior of the house, and are supported in the middle of their length, by the columns J, J. On these beams, also, the flooring of the lower and upper flat is disposed.

The fly-wheel K, K, of the steam engine is of that kind denominated *Spur fly-wheels*, from the circumstance of their being formed with teeth on the exterior of the rim, and thus serving at once to regulate the velocity, and to transmit the power of the steam engine to which they are attached. The introduction of this method, which is now very extensively practised, is also due to Mr. Fairbairn. The spur fly-wheel K, K, the diameter of which is 9 feet 3¾ in., geers with the pinion L, of 4 feet 10¾ in. diameter; consequently, the velocity of the crank-shaft is nearly doubled upon the horizontal shaft M, M, to which the latter is fixed, and which, by means of the bevil wheels and pinions N, N, N, gives motion to the stones contained within the mill-stone cases P, P, P. The shaft M, M, has a bearing upon the wall E, close to the back of the pinion L, and one in each of the standards O, O, O, to which the mechanism necessary for impelling and regulating the action of the stones is attached. This mechanism will be very fully detailed and described hereafter, and need not here be further adverted to.

Considerable difference of opinion exists among the millwrights of the present day regarding the comparative advantages of spur* and bevil geering as employed for driving grinding machinery. Into this question our limits do not admit of our entering; in our examples we have chosen the latter method, as being that most generally practised. We may, however, be permitted to enumerate a few of the more obvious advantages attending the present system. 1st, It admits of the stones, whatever may be the number employed, being ranged in a straight line instead of in a circle, thereby economising space and tending to a more convenient and economical disposition of the garners and apparatus by which they are fed. 2nd, It dispenses with the cumbrous and expensive frame-work necessary for binding together the parts of the system of spur-geering. 3dly, It admits of the employment of wheel-work of a finer pitch, and consequently of a more smooth and equable action, than could be used in the other case. And, 4thly, The use of bevil geering increases the facility of disengaging at pleasure any pair of stones which may require examination or repair.

We shall now proceed to describe the mechanism by which the various processes undergone by the grain, both previously and subsequently to the grinding, are effected; and, to avoid repetitions, we shall notice these processes in the order in which they occur.

The corn to be ground is deposited in the upper floor of the mill in the large garner Q, Q, (fig. 3), from which it is conducted through the spout R, into the *Screening* machine S, S, where it is cleansed of the dust and other extraneous matter which is found more or less combined with it. This machine consists of a species of cylindrical sieve, formed of wire-cloth, and partitioned inside so as to resemble an Archimedian screw. It is mounted upon an axis, and revolves with a considerable velocity in the interior of a close box, in which it is set at an angle with the horizon. The corn enters at its upper extremity, and, after being thoroughly agitated by its passage through the partitions in the interior of the screen, and thereby divested of the greater portion of the refuse with which it was mixed, falls into a spout U, at its lower end, which conducts it to the *elevator* V; being subjected, in its passage through this spout, to the action of a blast from the fan T, by which the remaining portion of the sand and dust that escapes with the grain is carried off by a passage leading to the exterior of the house. The grain, after being thus cleansed, is caught up by the elevator V, and raised nearly to the summit of the mill, where it is delivered, through an inclined spout X, into the *Creeper-box* Y, Y, by which it is distributed into the feeding garners Z, Z, Z.

The *Elevator* consists of a long endless chain of small buckets formed of tin-plate, and mounted at regular distances upon a leather band passing over two pulleys inclosed within the cast-iron frames V, V. The uppermost of these pulleys is driven at a moderate velocity by a belt, and the buckets, passing in succession the opening by which the grain is delivered from the screen, become each charged with a small portion of it; this they convey through the wooden pipes or boxes W, W, in which they are inclosed, to the upper extremity of the chain, where they deliver their contents into the spout X.

The contrivance just described is applicable only to the raising of the grain or flower from a lower level to a higher. For horizontal transport, modern millwrights make use of an apparatus called the *Creeper*, which is a very happy application of a well known principle, to the abridgment of manual labour. The creeper is simply a long endless screw with a wide pitch and thin threads, inclosed in a wooden box or trough, of dimensions slightly greater than its own diameter. It is made to revolve upon its axis, by means of a belt and pulleys, at a velocity corresponding with that of the elevators, and, being restricted from moving longitudinally, the threads, or rather leaves, of the screw force the grain introduced at one end of the trough to the other. The action of the screw in the case of the creeper is identical in its nature with that of the endless screw in giving motion to a worm-wheel. We shall have occasion subsequently to describe minutely the form and construction of the creeper employed by Mr. Fairbairn, and need only, in reference to the present Plates, to point to the views given of it in figs. 1 and 2.

The corn which is supplied to the garners Z, Z, Z, falls through the feeding pipes, or spouts A′, A′, A′, into the hoppers, by which the grinding apparatus is surmounted. After being reduced into flour, it falls through the pipes B′, B′, B′, into the creeper-box Y′, Y′, by which it is transferred to the elevator V. By this elevator it is again raised to the summit of the house, and carried by means of the creeper Y′, to the *Dressing* machine S′.*

This machine, which is very similar in external appearance to the screening machine already described, consists of a hollow cylinder covered with wire-cloth of different degrees of fineness, the finest being at the end which is most elevated. Within the cylinder, which is stationary, a circular brush revolves, in contact with the wire-cloth of which it is composed. The flour which is fed into the cylinder is, by the motion of the brush, sifted or rubbed through the wire, the finest through the upper end, the second through the next division, and so on, till the bran falls though the end of the cylinder, being too coarse to pass through any of the wires.* The different products thus separated are then stored in sacks, or otherwise disposed of as may be most convenient.

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* When spur geering is employed the arrangement is as follows: a great spur wheel, fixed upon a vertical shaft driven by bevil geering from the prime mover, revolves in the centre of a system of stones, the number of which rarely exceed six; each of these is driven by a pinion geering into the great spur-wheel, which thus commands the whole simultaneously.

* It may be remarked that, besides the great saving in manual labour effected by the use of these contrivances for transporting the flour from one part of the mill to another, their use is accompanied with another very important advantage: by exposing the flour liberally to contact with the air, the heat evolved during the process of grinding is abstracted, and the moisture evaporated; both which operations it is of essential importance that it should undergo previous to the process of dressing.

The sack-tackle is very simple, and will be readily understood from the sketch given of it in fig. 3. It consists of a barrel H′, provided with a rope of sufficient length to reach to the lower floor of the mill, and fitted to revolve in bearings attached to the roof; it receives motion, when required to be brought into action, from a belt connecting the pulley on its axis with a shaft G′, worked by the engine. The length of this belt is so adjusted that the sack-tackle may remain at rest, or be set in motion, according as the long lever I′, (the action of which is to tighten or relax the belt as may be required), is drawn to the right or left. In the floors of the mill are formed square hatches or openings, K′, K′, through which sacks, &c., may be admitted; and the rope of the sack-tackle passing over a guide pulley situated immediately above the centre of these hatches, thus affords a ready means of raising any heavy articles to either of the flats of the mill.

The geering by which the subordinate machinery of the mill is driven consists, first, of an upright shaft C′, C′, set in motion by a pair of bevil wheels *a*, from the main horizontal shaft M, M. This shaft has its lower bearing in an arched standard embracing the shaft M, and at its upper extremity it is supported by a plummer-block bolted to a double bracket D′, embedded in the wall E. Its motion is here transferred by means of another pair of bevil wheels *b*, to the horizontal shaft E′, E′, passing transversely across the mill. On this shaft are fixed the pulleys *d*, and *e*, which drive the screening and dressing machines respectively, and a set of small bevil wheels *c*, *c*, serve to transmit the motion to the longitudinal shafts F′, F′, by which the elevators, creepers, &c. are propelled; as also the short shaft G′, by which the sack-tackle is driven.

The dimensions of the various wheels, pinions and pulleys employed in this mill, and the velocities imparted by them to the machines driven by them respectively, are given in a tabular form at the head of next column.

Steam Engine F, 12 Horse Power, makes 40 Revolutions of Crank-shaft per Minute.

Description of Geering.	Driver.			Driven.		Result. Revolutions per Minute.	
	Diameter. Ft.	Ins.	Revolutions.	Diameter. Ft.	Ins.		
Spur Pair K, L,....	9	3¾	40	4	10¾	76	on horizontal shaft M.
Bevil Pairs N, N, N,	3	6	76	1	10	140	on the stones.
Bevil Pair *a*,.........	3	6	76	1	10	140	on upright shaft C′.
Bevil Pair *b*,.........	3	0	140	1	9	242	on transverse shaft E′.
Bevil Pairs *c*, *c*,......	1	1½	242	1	11½	140	on longitudinal shafts F′, F′.
Pulley *d*,..............	1	6	242	1	0	363	on screening machine S′.
Pulley *e*,..............	1	6	242	1	0	363	on dressing machine S′.
Pulley *f*,..............	2	0	140	0	6	560	on fan T.
Pulley *g*,..............	0	8	140	2	0	46·6	on elevators and creepers.
Pulley *h*,..............	1	0	140	2	0	70	on intermediate shaft G′.
Pulley *i*,..............	1	6	70	2	0	47	on sack-tackle H′.

Literal References.

A, A, A, the sheet-iron sides of the house in which the mill is erected.
B, B, B, columns for supporting and strengthening the structure.
C, C, horizontal beams at the level of the first floor.
D, D, the roof, formed of corrugated iron.
E, E, a wall of mason-work, affording a foundation for the bearings of the driving geering, &c.
F, the steam engine by which the mill is driven.
G, G, the boilers.
H, H, the flues and seat of the boilers.
I, I, I, transverse cast-iron beams for supporting the floors, &c.
J, J, columns for supporting these beams.
K, K, the spur fly-wheel of the engine.
L, pinion working into the above.
M, M, the main horizontal shaft.
N, N, N, bevil mortice wheels and pinions by which the millstone spindles are driven.
O, O, O, the standards of the grinding machinery.
P, P, P, the millstone cases.
Q, Q, the large garner for uncleaned wheat.
R, spout leading from the garner Q to
S, the screening machine.
S′, the dressing machine.
T, a fan attached to the wheat screen.
U, spout leading from the wheat screen to
V, W, the first elevator.
X, passage conducting the grain from the first elevator to
Y, Y, Y, the creeper by which it is distributed into
Z, Z, Z, the garners for feeding the stones.
A′, A′, A′, the feeding pipes.
B′, B′, B′, pipes by which the flour is delivered into
Y′, Y′, the second creeper-box, conducting it to
V′, W′, the second elevator.
a, bevil wheel and pinion giving motion to
C′, C′, the vertical shaft of the mill.
D′, a cast-iron support for the bearings of the vertical and transverse horizontal shafts.
b, bevil wheel and pinion giving motion to
E′, E′, the transverse horizontal shaft.
c, *c*, a set of small bevil geering, giving motion to
F′, F′, the longitudinal horizontal shaft.
d, *e*, *f*, *g*, *h*, *i*, pulleys for giving motion to the various subordinate machinery of the mill.
G′, intermediate shaft, conveying motion to
H′, H′, the sack-tackle.
I′, a lever for starting and stopping the sack-tackle.
K′, K′, hatchways by which the sacks are admitted or withdrawn.

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* On the processes undergone by the corn both previously and subsequently to the grinding, much of the success of the whole operation depends. In place of the wheat screen, an apparatus called a *Sheeling mill* is employed in some establishments. This consists of a pair of ungrooved millstones working at such a distance apart that the grain is merely rubbed between them, but not cut or broken. From the stones it is received upon an inclined sieve, where the heavier parts of the refuse fall from it, and is then exposed to the blast of a fan, which deprives it of the remaining lighter portions. For dressing the *flour*, *Bolting* machines are very generally used either in combination with, or in place of, the dressing machines described above. The determination of the best means of effecting these subordinate processes comes, however, more within the province of the *miller* than of the *millwright*.

PLATES CII.—CV.

DETAILS OF CORN MILL. BY WILLIAM FAIRBAIRN, MANCHESTER.

These Plates exhibit very fully the constructive details of the grinding apparatus, and of some of the subordinate machinery employed by Mr. Fairbairn in all his most recently erected mills, whatever may be their extent, and the number of stones they contain. The height of the *stone floor*, (or level at which the millstones are situated), from the foundation, varies slightly in different mills, and the diameter of the driving shaft must be increased or diminished in proportion to the number of pairs of stones to be driven, and to the distance of each individual pair from the prime mover. The only deviations from the form and dimensions here given, are, in the former case, to alter the length of that part of the framing called the Cone, B, B, and in the latter, to increase or diminish the diameter of the bearings in the plummer-block G, G.

These drawings are carefully laid down according to the different scales appended to them, and to render them of more value to the practical machinist, the dimensions of the principal parts are marked upon them.

Enumeration of the Plates.

Plates CII., CIII.—Fig. 1 is a front elevation of the grinding machinery, exhibiting the external appearance of the whole.

Fig. 2 is a general section taken on a vertical plane at right angles to that of the elevation, (fig. 1). In this view the millstones and the whole grinding apparatus are shown in combination.

Fig. 3 represents a plan of the cone, showing the number and disposition of the adjusting screws of the lower stone.

Fig. 4. No. 1. An enlarged external view of the feeding tube; No. 2, an enlarged elevation, and No. 3, a plan of the lever for regulating the supply of grain admitted to the stones.

Fig. 5. A plan of the ring forming the top of the tripod on which the feeding hopper is supported.

Fig. 6. A plan of the ring by which the adjusting wedges of the upper bearing of the mill-spindle are regulated.

Fig. 7. An enlarged view of the mill-spindle.

Fig. 8. No. 1. An external elevation; No. 2, a vertical section; and No. 3, a plan of the *Rhind*, or cross-head by which the upper stone, or *runner*, is connected to the spindle, and the grain supplied to the stones. No. 4 is a face view, and No. 5, a plan of the sockets in which the rhind works.

Fig. 9 is a plan of one of the millstones, showing the disposition of, and mode of drawing, the grooves on its surface.

Fig. 10. No. 1 is a plan of the *bed-stone box*, or cast-iron socket inserted into the lower stone, and forming the upper bearing of the mill spindle; No. 2, an edge view, and No. 3, a front view of the brass bushes fitted into this socket.

Fig. 11. No. 1. A face view, and No. 2, an edge view of one of the wedges by which the brasses of the mill-spindle are adjusted.

Fig. 12 is a sectional plan of the principal standard or frame of the mill, showing the mode of adjusting the lower end of the mill-spindle. The line of section is taken on the dotted line 1—2 in the general section, (fig. 2).

Fig. 13 is a sectional plan of the standard taken on the line 3—4, (fig. 2), and exhibiting the mechanism for disengaging the driving pinion when necessary.

Plates CIV., CV.—Fig. 14 is a partial section of the standard and attached mechanism. This section is taken on a plane corresponding with that of fig. 1.

Fig. 15. No. 1, is a side elevation, and No. 2, a plan of the plummer-block of the driving shaft.

Fig. 16. No. 1. An external elevation, and No. 2, a plan of the driving pinion of the mill.

Fig. 17. No. 1. A side view, and No. 2, a plan of the lever for adjusting the level of the upper stone.

Fig. 18. No. 1. is an edge view; No. 2, a front view of the screw and nut by which the adjusting lever is regulated.

Fig. 19. No. 1. An edge view; No. 2, a plan of the key used for working the above adjusting apparatus.

Fig. 20. The figures given under this head represent the different subordinate parts of the mechanism by which the lower end of the mill-spindle is supported, and its position adjusted.

Fig. 21. These figures represent the details of the mechanism employed for raising the driving pinion out of geer with the wheel.

Fig. 22. No. 1, is an external elevation; No. 2, a vertical section; and No. 3, a plan, on an enlarged scale, of the small apparatus used for regulating the supply of grain to the mills.

Figs. 23, 24, and 25. These figures give a representation, on a large scale, of the various adjusting screws required in this grinding machinery: the first being that for adjusting the lower end of the mill-spindle; the second for centering, and the third for levelling the lower or fixed stone.

Fig. 26. These figures represent general and detailed views of a species of small portable crane or jack, as made by Mr. Fairbairn, and employed for the purpose of raising and replacing the upper millstones when they require to be re-dressed.

Fig. 27. No. 1, is a side elevation, and No. 2, an edge view of the lower elevator framing.

Fig. 28. These figures give minute representations of the construction of the creeper employed by Mr. Fairbairn. Nos. 1 and 2 are a section and external view combined, to show the mode of junction of the different lengths; No. 3, is a tranverse section of the creeper box; No. 4, a plan of the plug or connecting block; No. 5, a view of the stud or journal which terminates the range, and No. 6, of that which commences it, and on which the driving pulley is keyed.

General Description.

The Framing.—A strong cast-iron standard or framing A, A, securely bolted to a stone foundation by two holding-down bolts, incloses the principal part of the driving and adjusting geering for each pair of stones. It is made in the form of an oblong box, and is traversed by two horizontal diaphragms or partitions, cast of a piece with it, the upper one for sustaining the foot-step of the mill-spindle with its adjusting apparatus, and the lower for carrying the plummer-block of the driving shaft. It is surmounted by a large bell-shaped casting B, B, called the *Cone*, firmly bolted, by a flange at its lower end, to the standard, while the upper extremity is expanded, and terminates in a cylinder, of a diameter somewhat greater than that of the millstones, the lower of which (sometimes called the *bed-stone*) rests, and is secured within it. Two straight and broad flanges are cast at opposite sides of the cylindrical part, for the purpose of bolting the cone to the beams of the mill, or to the same parts of the framing of the contiguous pairs of stones; while another circular flange passes all round, for sustaining the flooring. Three large openings are left in the upper part of the cone to give access to the interior, and it is provided with suitable arrangements for the reception of the several adjusting screws required for the setting of the lower stone.

The Stone-Case and Feeding Hopper.—Above the cone, and of the same diameter with the cylindrical part of it, is placed the stone-case C, which surrounds the upper stone, and serves to confine the flour which is the result of the grinding. This is simply a cylinder of thin sheet-iron, resting upon the stone floor, and having affixed to the top of it a ring of wood, on which the tripod for supporting the feeding apparatus is set. This cover is made open in order to admit the air freely between and around the stones during the process of grinding. A cast-iron ring D, supported by three malleable iron legs *a*, *a*, *a*, forms a sort of tripod in which is placed the hopper E, which receives the grain from the garners above, through the feeding pipe or spout *b*, and supplies it to the stones by means of the feeding apparatus to be hereafter described. A piece of coarse wire gauze is placed in the hopper, to intercept any foreign body that may descend with the grain.

The Driving Geer.—The driving shaft F is part of the line of horizontal shafting which is common to the whole range, and which receives its motion from the prime mover, generally through the intervention of a single pair of wheels. The velocity of this line of shafting is usually from 70 to 80 revolutions per minute, with stones of the diameter of those in our examples. The different lengths of which it is composed are connected together by couplings of the same description as that described and represented at fig. 15, page 143 of this work. The shaft F revolves in brass bearings, fitted into a plummer-block G, bolted to a sole formed, as before noticed, in the standard A. The strain of the shaft being entirely in a downward direction, this plummer-block requires no cover, the journal being simply protected from injury by a slight brass cap.

A large bevil mortice wheel H, working into the pinion I, on the mill-spindle, serves to transmit the motion of the shaft F to the latter. These wheels are made with the greatest possible care and accuracy, so as to work together very smoothly. The pinion is not fixed immovably upon the spindle, but is capable of sliding vertically upon it by means of a sunk feather.

The Mill-Spindle and its Appendages.—The mill-spindle J, J, is made of the best forged iron, accurately turned over its entire length; and rises perpendicularly through the standard A, the cone B, and the lower millstone. It is attached to the upper or running stone by means of a cast-iron piece K, called the *Rhind*, which combines this function with that of regulating and delivering the supply of grain to the stones. It is in the form given to this very important part, that Mr. Fairbairn's most recent improvement in grinding machinery consists. It will be observed by the drawings (fig. 8), that it forms a species of universal joint; the small steel cross-head *c*, *c*, on the top of the mill-spindle, fitting into corresponding bearings in the rhind, while the projecting tails *d*, *d*, cast upon it at right angles to the former, work in similar bearings formed of small cast-iron pieces sunk into the stone. By this arrangement it will be observed, that the connection between the mill-spindle and the upper stone is complete, while at the same time it admits of the free and unconstrained action of the latter against the grinding surface of the lower stone.

The lower, or fixed stone is perforated by a large square hole in its centre, into which the cast-iron block L (fig. 10), is firmly fixed by slips of wood and wedges. Into this block are fitted the three brass bushes *e*, *e*, *e*, which form the upper bearing of the mill-spindle. These are adjusted by means of the wedges *f*, *f*, *f*, the screwed tails of which pass downwards through the cast-iron ring *g*, and are regulated by thumb-screws on each side of it. The large openings in the cone, before alluded to, afford access for the working of these screws. Small semi-circular chambers are formed in the socket L, between each bush, and filled with hemp and tallow, for the lubrication of the mill-spindle; and the whole is carefully protected from dust by slips of sheet-iron screwed over it, (see fig. 9.)

The Millstones.—The diameter of the millstones most in use at the present day is 4 feet, and their thickness about 12 inches; one half of this thickness is composed of French burr, a very hard, though porous mineral, of a silicious nature; the other half is made up of plaster of Paris. In consequence of the difficulty of obtaining sufficiently large masses of the French stone, it is usual to construct the millstones in segments, which are cemented together, and the whole firmly bound by iron hoops passing round the circumference. The lower stone is, in the first instance, carefully dressed into a perfectly flat, plane surface, but the upper one is made slightly hollow for a small distance from the central aperture, so as to allow the grain to be freely admitted between the stones. Being thus prepared, grooves are then cut on the rubbing surfaces of both, in the manner indicated in fig. 9. The circumference of the stone is first divided into eleven equal parts;* lines are drawn from each division to the centre; these radii determine the limits of the grooves in each compartment. A chord $b' c'$, is then drawn, joining the bounding radii of any two compartments; this chord is, of course, bisected by the intermediate radius

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* The number of channels formed in the stones, and consequently the number of compartments, or *quarters* into which they are primarily divided, are varied by different millers, but the mode of drawing the lines, as here given, is applicable in all cases.

J a', in d'. Divide the line $d'\,c'$ into four equal parts in the points e', f', g'; and from these points mark off, on the line $d'\,c'$, distances equal to the width of the groove to be cut; then draw through all these points of division lines parallel to the radius J a', terminating them in the radius J c'. These are the outlines of the grooves, which are then to be cut into the stone, perpendicularly on one side and obliquely on the other, so that each furrow shall have a sharp edge. The direction of the grooves being the same in both upper and lower stones, as they lie on their backs in the position proper for being cut, it is obvious that when the former is reversed and set in motion, their sharp edges will meet each other after the manner of a pair of scissors, (as partially shown by the dotted lines in fig. 9), and thus grind the corn more effectually when it is subjected to the action of the unbroken surfaces between the channels.

Adjustment of the Lower Stone.—It is of the most essential importance to the proper working of any pair of stones, that the grinding surface of the lower stone should be perfectly level, and that its centre should be exactly perpendicular above that of the lower bearing of the mill-spindle. To secure the former of these conditions, three pinching screws *h*, *h*, *h*, are fitted into the cone, (that number being greatly preferable to four in adjusting the *level* of any surface), and, bearing against small slips of iron sunk into the stone, it can be raised or depressed by them to any required extent. The *centering* of the stone is effected by means of *four* pinching screws *i*, *i*, *i*, *i*, acting horizontally upon it. To secure it against deviating from the truth after having been properly adjusted, all these screws are provided with jam-nuts.

Adjustment of the Mill-Spindle.—The lower bearing or foot-step of the spindle J is also made capable of nice adjustment, both horizontally and vertically. The former is necessary in order to ensure the accurate working of the driving wheel and pinion, and the latter to regulate the pressure of the upper upon the lower stone, and to compensate for the changes produced upon both by the frequent dressings which their grinding surfaces have to undergo.

The footstep *k*, which is of gun-metal, is turned and fitted accurately into a cast-iron socket *l*, resting on the upper diaphragm of the standard A; the hole into which it is inserted and the annular recess by which it is surrounded being made of somewhat greater diameter than the corresponding parts of the socket itself. Its exact position is determined and secured by the four radial pinching screws *m*, *m*, passing through the ring and working in nuts fitted into recesses cast upon its interior surface, (see figs. 12 and 23). The footstep *k*, is not fixed immovably into the socket *l*, but is capable of sliding vertically in it. Its proper position, in this direction, is regulated by means of a strong wrought-iron lever M, having its centre of motion in the back of the standard A, while its opposite end projects through a slot, and is raised or depressed by means of a screwed rod N, jointed to it, and passing through a projecting shelf cast upon the front of the standard. A small link or saddle *n*, serves to connect the lever with the footstep *k*; the saddle being provided with a *square* tail which is inserted into a similar recess in the under side of the footstep; by which means the latter is prevented from turning in its socket. Thus it will be seen that the entire weight of the upper stone and mill-spindle rests upon the lever M, and that the miller is enabled to vary at pleasure the pressure upon the grain between the stones, and consequently, the degree of fineness of the flour produced, by simply turning the nut of the screw N by means of the key *o*.

The Feeding Apparatus.—The supply of grain admitted between the stones is regulated by means of a cast-iron pipe O, open at both ends, the lower end being brought into close proximity with the rhind, while the upper part incloses the pipe in which the feeding hopper E terminates. It is suspended by means of a cast-iron lever P, which has its fulcrum in the small column *p*, depending from the tripod D. A small chain attached to the end of the lever, and passing over a friction pulley at the bottom of the stone-case, serves to connect this feeding apparatus with an ingenious little piece of mechanism Q, attached to the standard, by which the miller is enabled to regulate the supply with the greatest nicety.

This contrivance, (which is represented on a large scale at fig. 22), consists of a small hand-wheel *q*, working between the cheeks of a double bracket bolted to the standard A. A small screwed pin forms the axis of this wheel; it passes freely through the cheeks of the bracket, but is screwed into the eye of the hand-wheel, and is prevented from turning with it by means of a feather inserted into the former, and fitting into a groove cut throughout the entire length of the pin, to the upper end of which the chain is attached. By this arrangement it is obvious that by turning the hand-wheel *q*, to the right or left, the small pin will be raised or depressed, and through the intervening mechanism, the size of the opening between the mouth of the feeding pipe O and the rhind will be increased or diminished.

The form given by Mr. Fairbairn to the upper surface of the rhind constitutes one of the chief peculiarities of his improved system of grinding machinery and is an admirable combination of simplicity and perfect efficiency. We have been at great pains to give an accurate representation of it in our various views, (see figs. 2 and 8). The principle by which its form is determined is, that the centrifugal force of the running stone shall be employed in the most effectual manner to distribute the corn equally over every point of its surface, while, at the same time, the supply admitted shall be susceptible of the nicest possible adjustment.

The Disengaging Apparatus.—The driving pinion I is fitted upon the mill-spindle so as to be capable of sliding up and down upon a sunk feather. When fully in geer with the wheel H, it rests upon a collar formed on the upper surface of a large brass nut *j*, (fitted to a screwed part on the lower end of the spindle, and capable of being fixed by a jam-nut, see fig. 7), by which the miller is enabled to keep the pinion invariably in its proper position with regard to the wheel, independently of the position of the spindle, which, as we have before had occasion to remark, requires to be slightly lowered every time the stones are dressed. When properly adjusted the pinion is secured to the spindle by a tapered key.

It is, however, necessary to throw each pair of stones, periodically, out of geer with the general range, to admit of their being dressed, &c. For this purpose the tapered key is removed, and the pinion raised out of contact with the teeth of its driving wheel, by means of a species of small jack or lifting apparatus attached to the standard, the component parts of which we shall now briefly enumerate. A cast-iron ring R, supported upon two upright rods *r*, *r*, is brought into contact with the under surface of the pinion by turning the hand-wheel *s*, which is screwed upon its axis *t*, and carries with it in its ascent, the cross-head S, into the ends of which the lower extremities of the rods *r*, *r*, are inserted. The screw *t*, is fixed into a socket cast upon the under surface of the lower diaphragm of the standard, and the connecting rods *r*, *r*, pass through holes formed for their reception in both diaphragms; being set in a diagonal direction in order to clear the lever M, and other important parts of the machinery.

On turning the hand-wheel in the contrary direction, the weight of the pinion again brings it into its working position.

The Stone Lifting Machine.—Fig. 26 is a representation of a portable lifting apparatus, constructed by Mr. Fairbairn, and extensively employed in the mills fitted up by him, for raising the upper stones from their beds and depositing them on the floor of the mill, when they require to undergo the process of dressing. It consists of a strong malleable iron arm T, T, bent into a form nearly approaching to a quadrant; the lower end works in a cast-iron step *v*, inserted into the stone floor, while its upper extremity is supported by a strong rod U, fitted to rotate upon a stud *u*, fixed into a cast-iron plate (No. 4), bolted to the beams which support the floor above. The fixed centres *u*, and *v*, are so situated that the machine shall command two contiguous pairs of stones, (see Plate CVI.), and the end of the rod U is made of such a form as to admit of its being easily disengaged from the stud; when the entire machine may be removed. A strong screw V, passing through the arm T, and worked by means of a nut formed into a double handle, carries at its lower end the two connecting links W, W, which are attached to the stone by two studs temporarily inserted into it at points diametrically opposite. The links W, W, are bent so as to admit of the stone being inverted while it is suspended in the lifting machine. The running stone is retained in its place in the mill, simply by its own weight; it is, therefore, only necessary to raise it out of its bearings when the grinding surfaces require examination or repair.

The Elevators and Creepers.—Fig. 27 is a representation of the form given by Mr. Fairbairn to the lower elevator-frame; the nature and construction of this machine having been fully noticed in our previous description, it is not necessary further to advert to them. The creeper, however, (fig. 28), as constructed by Mr. Fairbairn, is an object of greater interest. He has replaced the rude, cumbrous and expensive contrivances of former millwrights, by a form at once lighter, cheaper, and greatly more efficient. The material employed is cast-iron; the creeper is made in 6 feet lengths; each length being in the form of a tube, $3\frac{1}{2}$ inches in diameter, and about $\frac{3}{8}$ of an inch thick, with broad leaves or threads cast round it after the manner of an Archimedian screw. The thickness of the threads does not exceed $\frac{3}{16}$ of an inch, at the outer extremity.

The different lengths of which the entire creeper is composed, are joined together by short wrought-iron studs *x*, *x*, forming also the journals on which it revolves. These are made with square tails fitted into similar holes formed in the centre of the small cylindrical blocks *w*, *w*, which are carefully turned on their exterior surfaces, and driven into the open ends of the pipes Y, Y, previously bored to the same diameter. This construction at once ensures a strictly rectilinear axis for the entire range, whatever may be its length. Minor details will be understood by a simple reference to the figures. The arrow indicates the direction in which a creeper constructed in the manner shown in the general view would propel the grain.

Literal References.

A, A, the standard or lower framing of the grinding machinery.
B, B, the cone or upper framing.
C, the stone case, of sheet-iron.
D, a cast-iron ring for supporting the hopper, carried upon
a, *a*, *a*, three wrought-iron legs, resting upon the top of the stone case.
E, the feeding hopper.
b, a pipe for supplying grain to the hopper from the garners above.
F, the main driving shaft.
G, plummer-block of the shaft F.
H, a bevil mortice wheel, conveying the motion of the shaft F to
I, the pinion of the mill-spindle.
J, the mill-spindle.
K, the rhind of cast-iron by which the spindle is connected with the upper stone.
c, *d*, the bearings of the universal joint formed by the rhind.
L, the bed-stone-box for the upper bearings of the mill-spindle.
e, *f*, bushes and wedges for adjusting the bearings.
g, a thin cast-iron plate, by means of which the wedges *f*, *f*, are adjusted and fixed.
h, *h*, *h*, pinching-screws for adjusting the level of the lower mill-stone.
i, *i*, *i*, *i*, pinching-screws for centering it.
j, a large brass nut for supporting the pinion I.
k, footstep of the mill-spindle.
l, cast-iron socket for the footstep *k*.
m, *m*, pinching-screws for adjusting the socket *l*.
n, the saddle or link connecting the footstep *k*, with
M, the great lever for supporting and adjusting the mill-spindle.
N, *o*, screwed rod and key for working the lever M.
O, the movable feeding pipe, of cast-iron.
P, lever for adjusting the pipe O.
p, small column forming the centre of motion of the lever P.
Q, *q*, apparatus for regulating the feed.
R, a cast-iron ring for raising the driving pinion out of geer with the wheel.
S, a cast-iron cross-head, being part of the same apparatus.
r, *r*, upright rods connecting the cross-head S with the ring R.
s, *s*, *t*, hand-wheel and screw for working the disengaging geer.
T, U, V, W, the several parts of which the stone-lifting machine is composed.
u, *v*, the centres of motion on which it turns.
X, the lower elevator frame.
Y, Y, the creeper, of cast-iron.
w, *x*, blocks and studs for connecting the adjacent lengths of the creeper.
y, the brackets in which it revolves.
Z, Z, the creeper-box, of wood.

PLATE CVI.

OLD UNION CORN MILL, BIRMINGHAM.

In order to render our series of Corn Mill drawings complete, we have given in this Plate a representation of the arrangement of the stones, garners, &c. in an extensive establishment fitted by Mr. Fairbairn with machinery identical, in all important particulars, with that which we have detailed in the preceding Plates and descriptions.

This mill contains at present twelve pairs of stones, but an addition to it is in course of erection, by which five more pairs will be added to the number. The whole are disposed in one straight line, close to the wall of the mill—which arrangement, on account of its undoubted superiority to any other, has induced us to select this example. The flour produced is, for the most part, consumed by an extensive baking establishment adjoining the premises, and belonging to the same proprietors.

For our present purpose, we have deemed it necessary to give only a sectional elevation, on a small scale, and of a somewhat sketchy nature. The line of section must not be supposed to be taken in one vertical plane, but to vary in order to exhibit most distinctly the different parts.

General Description.

Two powerful steam engines are employed to drive the machinery of this mill; one of these communicates its motion to the line of horizontal shafting, near the end of the range, by means of a great bevil wheel A, keyed to the crankshaft, and working into the bevil mortice pinion B upon the mill shaft; the other, by means of a spur fly-wheel working into the mortice pinion C, situated towards the centre of the range. By thus applying the power at two points, the weight of shafting, and consequent loss from friction, &c. is materially diminished.

From the driving shaft D, D, D, the power is communicated by the bevil pairs *a, a, a*, to the mill-spindles, which are provided with suitable bearings and adjusting geer, in all respects identical with that we have already described. Between the first and second pairs of stones, one of the standards E, is employed for the purpose, at once of giving great stability and firmness to the shaft D, at the point where it receives the strain of the first engine, and of affording the means of transmitting its motion to the upright shaft *i*, driving the lighter machinery of the mill. A strong cast-iron bracket *b*, bolted through the wall of the mill, abuts against the framing, and the intermediate stays *c, c*, bind all three together laterally.

Each of the feeding hoppers *d, d, d*, is supported by a light cast-iron framing *e*, resting upon slender columns; the tripodal form is, however, preferable to this.

The feeding pipes H, H, H, are of tinned plate, so constructed as to be easily withdrawn from their connection with the wheat garners, when the stones are taken up; and at the top of each is a small valve or slide *h*, to shut off the supply of grain. Above these points the passage of the grain from the cleaned wheat garners is effected by means of wooden spouts I, I, I, converging to, and communicating with them.

The clean wheat garners J, J, J, are three in number, and are situated on the top floor of the mill, immediately over the stones. Between, and in front of these, are the garners K, K, which receive the wheat as it is first brought to the mill, and deliver it through numerous spouts or openings in the floor, to the great uncleaned wheat garners L, L, situated on the machine flat. From these the undressed wheat is conveyed, by means of a creeper and elevator, into the screen M, whence it falls through the spout P into an elevator on the stone floor, by which it is again raised to the upper story and distributed, by another creeper, into the cleaned wheat garners J, J, J.

The vertical shaft *i i*, is continued upwards till it terminates near the roof of the machine floor. Then its motion is transferred by means of a short transverse shaft, (which we have purposely omitted in our drawing for the sake of simplicity), and two pairs of bevil wheels, to the long horizontal range of shafting *k, k*, on which are fixed numerous pulleys for driving the various subordinate machinery of the mill.

The Pulley *l*, drives the lower creeper.
— *m* — the upper do.
— *n* — the sack-tackle.
— *o* — the first elevator.
— *p* — the second do.
— *q* — the fan, } belts working over
— *r* — the wheat screen, } guide pulleys.

The dressing and bolting machines are driven by pulleys on the transverse shaft. The apparatus Q for raising the stones is shown here in action; it may be remarked that it is found necessary, in this mill, to keep a couple of stone-dressers in constant employment; each pair of stones being, in succession, thrown out of geer for twenty-fours hours, to undergo the process of re-dressing.

CALCULATION OF VELOCITIES.

1st Steam Engine makes 48 Revolutions of Crank-shaft per Minute.
2nd do. 22 do. do. do.

Description of Geering.	Driver.			Driven.		Result. Revolutions per Minute.
	Diameter.		Revols.	Diameter.		
	Ft.	Ins.		Ft.	Ins.	
Bevil pair A, B,......	6	8¾	48	4	6	71·57 on horizontal shaft D.
Spurfly-wheel into C,	18	2⅜	22	5	7	
Bevil pairs *a, a, a,*	3	6	71·57	1	10½	132 on the mill-stones and upright shaft *i, i.*
1st pair of small bevils, ...	3	0	132	2	1½	184 on the transverse shaft.
2nd do. do.	1	6	184	1	10	149 on the longitudinal shaft *k.*

Diameter of the mill-stones, 4 feet.

Literal References.

A, bevil wheel on the crank-shaft of first engine.
B, the pinion on the mill-shaft working into the above.
C, a spur cog pinion on the mill-shaft, driven by the spur fly-wheel of the second engine.
D, D, D, the range of mill-shafting.
E, E, E, the standards or lower framing of the mills.
F, F, F, the cones or upper framing of do.
G, G, G, the stone cases.
a, a, a, the bevil wheels and pinions for driving the mill-spindles.
b, a strong bracket for strengthening the bearing of the pinion B.

c, c, cross stays for the same purpose.
d, d, d, the feeding hoppers.
e, e, e, the frames supporting the feeding hoppers.
f, f, f, the cast-iron adjustable feeding pipes.
g, g, g, small screw and hand-wheel for working the feeding apparatus.
H, H, H, sheet-iron feeding pipes from the machine floor to the hoppers.
h, h, h, slides for shutting off the supply of wheat when necessary.
I, I, I, wooden spouts or square pipes for leading the grain to the feeding pipes H, H, H, from
J, J, J, the cleaned wheat garners on the top floor of the mill.
K, K, the first uncleaned wheat garners, emptying their contents into
L, L, L, the great uncleaned wheat garners on the machine floor.
M, the brusher or wheat screen.
N, the fan attached to the wheat screen.
O, passage of the blast, carrying the dust, &c. outside of the mill.
P, passage of the cleaned wheat to the second elevator.
i, i, the upright shaft for driving the light machinery of the mill.
j, a pair of bevils between the upright shaft and
k, k, the range of longitudinal shafting down the mill.
l, m, n, o, p, q, r, s, pulleys for driving the various light machinery.
Q, the stone-lifting machine shown in operation.
t, t, t, studs affixed to the beams, affording fixed centres for the working of the lifting machine.

PLATE CVII.

MACHINE FOR DRESSING MILLSTONES.

We have repeatedly had occasion to notice, in our descriptions of corn-grinding machinery, the frequency with which it is found necessary to renew the grinding surfaces of the stones, and the great care and accuracy to be observed in thus preparing them for their work. The operation consists in roughening the surface of the stone, when worn smooth, by cutting delicate and very regular furrows between each of the radial grooves and parallel to them, and depressing the latter to a degree corresponding with the amount of the grinding surface removed. Skilful and experienced workmen are employed for this service, which they accomplish by the help of hammers, provided with chisel-edges of well-tempered cast-steel.

Simple as this operation may appear, the attempts hitherto made to perform it entirely by machinery have proved abortive; and we need not be surprised at this, when we consider the various degrees of hardness and compactness presented by different parts of the same stone, as well as the differences in the nature of the grain which is to come under its action. But although it may safely be pronounced impossible to dispense with manual labour in this process, it may be accomplished with much greater despatch and precision by the use of the simple machine, or more properly *instrument*, which we have represented in the present Plate.

This instrument serves simply to guide the hand of the workman, and to restrict the cutting edge of the hammer into a perfectly parallel and rectilinear course; the force and number of the blows being left entirely to the discretion of the workman. By a simple contrivance, it is also made available for regulating with the greatest nicety the degree of fineness, or in other words, the distance between the furrows, of the grinding surface of the stones, according to the nature of the work on which they are to be employed.

Enumeration of the Figures.

Fig. 1 is a side elevation; fig. 2, an end view, and fig. 3, a plan of the entire machine.

Figs. 4 and 5 are views of the hammers and cutting tool, detached from the machine.

Figs. 6 and 8, a sectional view and corresponding plan of the axis E, and its screw and nut.

Figs. 7 and 9, detached views of the saddle or guide for the hammer.

Figs. 10, 11, and 12, represent the various handles used for varying the distance between the cuts; the first having five, the second six, and the third four rays or points.

General Description.

The base of this machine, which simply rests without being fixed upon the dressed surface of the millstone A, consists of a cast-iron bar or plate B, also dressed on its under surface, and of sufficient weight to retain it in the position in which it is set, without at the same time impairing its portable character. On this sole-plate a carriage C is fitted to traverse longitudinally; being supported at one side by a smooth cylindrical rod D, D, bolted to the sole-plate, and passing through the brass-mounted sockets *a, a*; while, at the other side, it is provided with small friction rollers *b, b*, resting on the sole-plate. At right angles to the direction of this motion, an axis E is fitted to slide on bearings cast upon the carriage C; the outer extremity of this axis is formed into a screw, worked by the nut F; and the opposite end, which is cranked, carries the saddle G, into which the hammer handle H is inserted, and which guides the action of the cutting tool. The nut F is elongated so as entirely to enclose the screw and protect it from injury; it is also provided with a set of star-handles c, c', c'', either of which may be affixed to it according to the degree of fineness which it is required to give to the stone. The first has six rays and gives the greatest degree of fineness, the distance between the cuts being only one-sixth of the pitch of the screw; the others, having fewer rays, give a correspondingly increased degree of roughness to the stone. A species of index *d*, fixed to the carriage, serves to regulate the exact amount of motion given to the nut. The cranked form given to the axis E is for the purpose of adapting the machine to the length of the cutting tool I. The manner in which this adjustment is accomplished will be obvious from inspection of the drawings.

Action of the Machine.—This instrument being laid upon the surface of the stone, with the edge of its sole plate exactly parallel to one of the great radial grooves, the carriage C is pushed by the hand of the workman towards the circumference of the stone, until the cutting tool I reaches it.

He then begins his strokes, varying the number and intensity of them according to the hardness of the stone, and the depth he wishes to give to the furrows; and drawing the carriage C slowly with his left hand towards the centre of the stone, until the tool reaches the inner boundary of the grinding surface. The hammer is then set into the position for making the next adjacent furrow, by turning the handle *c*, through another of its divisions; the carriage is again slid towards the circumference, and the operation continued as before. Thus the requisite straightness, regularity and parallelism of the grooves is ensured, and the operation of dressing materially facilitated.

Literal References.

A, the mill-stone to be re-cut.
B, the base-plate of the machine.
C, the carriage or support of the working parts.
a, *a*, straps of iron, binding the carriage to the cylindrical rod D.
b, *b*, friction rollers between the carriage and sole-plate.
D, D, a cylindrical horizontal guide-rod for the carriage C.
E, an axis sliding transversely in bearings, and carrying the guide for the hammer.
F, a nut for regulating the position of the axis E.
c, *c*′, *c*″, star-handles for regulating the fineness of the cut.
d, index for showing the amount of motion given to the nut F.
G, a cast-iron saddle serving as a guide to the hammer.
H, the wooden handle of the hammer.
I, the chisel or cutting tool inserted into the head of the hammer.

PLATES CVIII., CIX.

GOVERNORS FOR STEAM-ENGINES, WATER-WHEELS, ETC.

The subject brought under consideration by the Plates now before us possesses peculiar interest for the practical engineer, not only on account of the importance of the object proposed by all automatic regulators of the power of prime movers, but for the ingenuity displayed in the various contrivances which have been employed to accomplish that object. The common centrifugal, or *conical pendulum* governor of Watt has, since its first introduction, undergone many modifications of greater or less importance, affecting the symmetry of its appearance and the economy of its construction, as well as rendering it applicable to a great variety of purposes. Admirably as it is found, under ordinary circumstances, to perform its functions, it is nevertheless open to certain very grave objections, and much scientific and mechanical ingenuity has been employed in devising schemes for obviating these inconveniences, or for altogether superseding its use, although hitherto none of these projects has been very generally adopted.

We have, accordingly, in the present Plates, brought together a pretty extensive collection of governors of various kinds, partly with the view of furnishing our readers with some good examples of different modifications of the centrifugal governor, and partly for the sake of instituting a comparison between it and a few of the most remarkable contrivances which have been proposed in its stead. Into the theory of the centrifugal governor, and its general mode of action, we shall not, in this place, enter farther than may be necessary for the drawing of these comparisons; that subject having been fully discussed in many popular treatises.

In Plate XC. we have given, in connection with our series of Plates of Water Wheels, a very excellent example of the centrifugal governor as applied to regulate the quantity of water admitted into the buckets of an overshot wheel. Here the nature of the operation renders it necessary that a considerable amount of power should be put forth, and accordingly we find that the governor is employed merely to bring into action, at the proper time, and in the proper direction, a power altogether independent of that exerted by itself. In the first three figures of the Plates now before us, we give an example of a form adapted to the converse circumstances. This governor may be taken in connection with our series of Plates of Corn Mills; an arrangement of this nature being employed in many establishments to warn the millers, in case the stones should, from any cause, exceed or fall short of their prescribed velocity. Here all the work required of the governor is to ring a bell, and to indicate upon a dial the velocity of rotation of the mill stones.

This mechanism consists of a vertical wrought-iron axis A, A, revolving in bearings B, B, bolted to the wall of the mill, and carrying towards its upper extemity, a pulley C, which receives motion from the main driving shaft. To this axis is fixed a brass socket *b*, to which are jointed the two flat arms *a*, *a*, terminated by the governor balls, and attached, about the middle of their length, by the two double links *c*, *c*, to the sliding socket *d*, made in halves and connected together by two small bolts. To this latter are also attached the two slender vertical rods *e*, *e*, which traverse the pulley C, and convey the action of the governor to a sliding disc D, provided with a projecting arm or catch, of such length as to come into contact, should the machinery exceed or fall short of its proper speed, with either of the two levers *f*, *f*, which have their common centre of motion in a short vertical axis, and are attached at their opposite ends by slender wires to two sockets mounted upon a horizontal axis E; each of these sockets carries a bell which, by the arrangement described, is rung when the catch on the disc D strikes either of the levers *f*, *f*.

To the sliding socket *d*, is fixed a forked rod, having one of its branches formed into teeth like a rack; this rack geers with a small pinion, (see fig. 3), carrying upon its axis an index which points out upon the graduated dial F, the speed at which the millstones are revolving. Thus, should the velocity of the prime mover relax, the vertical axis A partaking of this diminished motion, the balls collapse, the socket *d*, is pressed downwards, and the rack causes the index to move from right to left. The opposite effect is produced by an increase of the speed; (these different positions being indicated by the dot lines in the figures). At the same time the bell is sounded by the apparatus which surmounts the governor; and the attendant, by a glance at the dial, is made aware of the change in the velocity of the machinery, for which he has to compensate by altering the degree of proximity of the upper and lower stones. It is sufficiently well known to our mechanical readers that the action of the governor is in no way affected by the weight of the balls, further than that these should be made of a size proportionate to the resistance to be overcome; accordingly, in the case now before us, the work which the governor is destined to perform being very slight, the balls may be made of extreme lightness.

Fig. 4 is an example of the original form of the governor as introduced by Watt and still extensively in use. It is taken from a French double cylinder or *Woolf* engine. The distinguishing peculiarity of this form of governor consists in the connecting links *c, c*,* being situated *overhead*, and attached to the arms *a, a*, by prolongations of the latter, which pass through a square part of the upright spindle A, to which they are both jointed by one pin. This form is wanting in compactness and elegance of appearance, and is, moreover, expensive and difficult of construction; it is, however, sufficiently convenient when the action of the governor is to be transmitted upwards. When at rest the balls are usually received into arms *g, g*, curved to suit their surfaces, by which means the rods are relieved from all unnecessary strain.

In fig. 5 we have given a representation of a very elegant and well-proportioned example of the centrifugal governor, adapted to a small high-pressure *crank-overhead* engine, by Messrs. James Gray and Co. of Glasgow. In this species of engine the governor is usually made to revolve in a short column B, cast in a piece with a forked bracket embracing the crank-shaft, and its spindle A, A, is driven by a pair of bevil wheels from the crank-shaft. The spindle is surmounted by a double brass socket *b*, attached to it by means of a pin; and to this socket are jointed the arms *a, a*, which, as well as the connecting links *c, c*, are, in this example, finished in the lathe. The sliding brass socket *d*, to which the lower ends of the links *c, c*, are connected, is formed with a groove, into which is inserted the forked end of a lever D, having its centre of motion in a small wrought-iron column bolted to an arm projecting from the column B. From the opposite end of this lever depends the slender rod *e*, connecting it immediately with the throttle-valve lever, which, by this simple construction, is at once made to rise or fall, as the balls collapse or diverge in obedience to the varying speed of the engine. This is a type of the most common form of the centrifugal governor in use at the present day.

Fig. 6 is an example of a very excellent arrangement of the conical pendulum governor, which is frequently adopted in highly-finished engines. The peculiarity of this form consists in the connecting rod *e*, being attached directly to the sliding socket *d*, without the intervention of the forked lever. For this purpose the upper portion of the spindle A, A, is bored out truly cylindrical, to a point somewhat below the range of the sliding socket *d*. This last is attached by means of a cotter to a small cylindrical hollow piece, which fits accurately into the interior of the spindle, and is consequently made to rise and fall with the socket *d*, a long slot being formed in the spindle to allow the cotter to traverse up and down. The lower end of the rod *e*, is jointed to this interior piece by means of a swivel, so as to rise and fall with it, without being affected by its rotatory motion; (see the small figure to the right of the principal one). At the top of the governor spindle, the rod *e*, is guided in its motion by being made to pass through the small brass vase which surmounts the whole apparatus, and should it be required to be of any considerable length, the necessary rigidity may be imparted by fixing a weight to it, as shown in the figure. This arrangement was first employed by Messrs. Hick and Rothwell of Bolton, to whom we are indebted for much of the elegance and beauty which characterises the modern Land Engine in all its details.

As in the example last under notice, the arms and links of this governor are finished in the lathe, as is also the double cup *g*, into which the balls are received when the engine is at rest. In this figure we have shown the balls in section, in order to exhibit the mode usually adopted for adjusting their position upon the arms *a, a*.

The form of governor represented by fig. 7, is that proposed by Mr. Henry Davies, the ingenious inventor and patentee of the Disc steam engine; and it appears to us to be worthy of the most favourable consideration, both on account of the principles on which it acts, and of the extreme simplicity and economy of its construction.

The action of the conical pendulum governor depends, as is well known, upon the combination, or rather on the conflicting influences, of the centrifugal force and that of gravitation; its parts being so proportioned that these forces shall be in equilibrium when the engine moves at its normal velocity; an increase or diminution of which causes the one force or the other to be in excess, and thereby brings the regulating power of the governor into action. A serious evil is found in practice to result from the very circumstance which theoretically constitutes the perfection of this contrivance: the extreme delicacy and sensibility of its action incessantly occasions an excess of motion in either direction, producing a series of alternations in the speed of the engine, which seldom settles down to a regular rate of motion, unless the production of steam and the burden upon the engine are, of themselves, either constantly uniform or proportional to each other; circumstances, be it observed, in which, supposing their occurrence to be possible, no governor would be required.

With the view of remedying these defects, the inventor of the governor now before us proposes to substitute for the invariable and uncontrollable force of gravity in the common centrifugal governor, an arrangement which, while it is not so easily affected by slight changes of velocity, shall yet be sufficiently delicate to compensate for all considerable variations, and shall, at the same time, be susceptible of the nicest adjustment to suit the varying circumstances in which an engine may be placed. For these purposes he employs a spiral spring as the resisting force to the centrifugal energy.

The vertical spindle A, A, which may be set in motion either by a pulley or by bevil wheels in the usual manner, is surmounted by two equal horizontal arms *a, a*, furnished with stops at their extremities. The governor balls run freely to and fro upon these arms by means of internal friction rollers, and are drawn towards the common centre in the spindle A, by means of cords or steel ribands *i, i*, passing over two pulleys at G, and attached at their lower ends to the sliding collar *d*, in which works the forked end of the lever D, D, for conveying the action of the governor to the throttle valve. A spiral spring embracing the vertical spindle presses at its lower extremity against the sliding collar *d*, and its pressure is regulated by a sliding stop *h*, which can be fixed at any required elevation upon the spindle by a set screw.

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* We have been careful to mark the corresponding parts in our various specimens of the centrifugal governor with the same letters; therefore, in referring to each, it is only necessary to point out its distinctive features.

The stop *h*, having been set so as to cause the spring to press down the collar *d*, with any approved force, according to the amount of load upon the engine, the latter is set in motion. Should its speed exceed the stipulated rate, the increased centrifugal force will cause the two balls to diverge, and raising the collar *d*, will partially close the throttle valve and diminish the supply of steam, when, the motion being checked, the spring will press down the collar, and cause the balls to collapse until the desired rate of motion is obtained.

The degree of force exerted by the spring will always require to be adjusted to suit the nature of the work thrown upon the engine, because a small quantity of steam only will be required when the work is light, and a larger quantity when it is heavy, while the speed in each case should be the same, which conditions can be fulfilled with great facility and admirable precision by the use of this kind of governor.

Fig. 8 represents a modification of the centrifugal governor, which, for simplicity of construction and compactness, appears to be deserving of especial notice. Here the balls, instead of being suspended upon arms, of a length proportioned to the velocity at which the engine is required to move, are fitted to traverse upon the arms H, H, which are fixed to, and consequently revolve with the spindle A, and are formed into circular arcs, of a curvature determined by the same circumstances with the length of the suspending arms in the ordinary governor. By this means it is obvious that the horizontal plane of the rotation of the balls will vary with the varying speed of the engine, in precisely the same way as in the conical pendulum governor; and the vertical motion thus generated is transferred directly to the sliding socket *d*, which commands the throttle valve lever. In order to effect this object, it is necessary that each of the balls should be made in halves, and riveted together with a wrought-iron pin, as shown in the section; a space being left between the hemispheres to admit of the slotted arms *a*, *a*, which are cast in a piece with the sliding socket *d*, and through which the connecting pins are fitted to pass freely, but without allowing any play.

This kind of governor has been fitted by Messrs. Tod and Macgregor of Glasgow, to several high-pressure engines constructed by them, and the result we understand to have been perfectly satisfactory.

The last variety of the centrifugal governor which remains to be noticed is that represented in figs. 9 and 10; the former being a sectional elevation, and the latter a plan of a governor constructed by M. Bourdon of Paris, the peculiarity of which consists in the axis of rotation being horizontal instead of vertical. The main advantage proposed to be attained by this system is the more convenient transmission of the motion of the prime mover, whether by wheel work or by pulleys. The principle of its action is the same as that of the common governor.

The spindle A, A is of cast-iron, the part to the left being hollow, while the middle portion is formed into a species of open frame-work, enclosing the principal part of the mechanism. It revolves in ordinary plummer-blocks B, B, and is set in motion by the cone pulley C. The arms *a*, *a*, which carry the governor balls, are supported upon a short axis working on the points of two steel pins, screwed into the central part of the spindle and secured by jam-nuts; this axis carries also a toothed sector *c*, working into a similar sector upon another short axis to which is fixed a lever *d*; the slender connecting rods *j*, *j*, traversing the hollow part of the spindle, and supported by the guides *k*, *k*, serve to convey the motion of this lever to the throttle valve geer, which is provided with suitable arrangements for adjusting the action of the governor upon the throttle valve.

In Plate CIX., we have grouped together a few of the most remarkable contrivances that have been proposed to obviate the defects and inconveniences that are found to attend the use of the conical pendulum governor, but before proceeding to a detailed explanation of them, we propose to give a brief enumeration of the gravest of these objections, in order that the reader may be enabled to judge of their validity, and of the pretensions of the several machines before us to superiority.

1st. Perhaps the most serious defect attending the use of the conical pendulum governor, as applied to steam engines, is that to which we have already had occasion to advert, namely, its liability to incessant oscillations or fluctuations of velocity, arising partly from its extreme delicacy of action, and partly from the slowness of its operation. It is obviously incapable of acting at the instant when the equilibrium between the power and the load is distributed, but only after the fly-wheel of the engine has received such impulse, that the governor balls revolve with sufficient velocity to overcome the resistance of the throttle valve, &c.

2nd. It is incapable of self-adjustment, as we have also noticed above. A different quantity of steam will be required for every *permanent* change of resistance, but the common governor can make no change in the opening of the throttle valve, except in consequence of a change of velocity in the engine. Consequently it is necessary, with every permanent increase or decrease of the load, to regulate the supply of steam by altering the length of the rods which connect the governor with the throttle valve, which, in many cases that will naturally present themselves to our readers, it is impossible or extremely inconvenient to accomplish.

3dly. From the exceedingly limited range of its action, the common governor is incapable of performing its functions in extreme cases. This we may illustrate by supposing an engine to be suddenly relieved of its load; the balls are consequently expanded to their utmost extent, but the amount of motion thereby generated in the parts which actuate the throttle valve is necessarily so small, as to be inadequate to avert the danger. Or, if the connections between the governor and throttle valve were such that, in the case we have supposed, the latter would be entirely closed, it is obvious that the same result must necessarily occur with a comparatively trifling increase of speed, say three or four revolutions above the normal velocity. To this defect we may add that the amount of power exerted by the ordinary governor is totally inadequate to the *direct* regulation of the shuttles of water-wheels, &c.; and that the intermediate geering required for such purpose tends greatly to enhance the inconvenience specified under our first head.

If to these defects we add the difficulty of calculating with sufficient accuracy the dimensions and velocity proper to be given to the conical pendulum governor to suit the circumstances in which it is destined to operate, and the extreme precision and nicety required in its construction, we shall not be at a loss to account for the multitude of contrivances which have been proposed to supersede its use.

The Air-Reservoir or Bellows Governor.—This apparatus is of French origin, a patent having been granted to the inventor M. Molinié, of Saint-Pons, in the department of Hérault, in 1838, for its application to water-wheels, and in 1840 to steam engines. Since the latter date it has been introduced into England, where, however, its claims do not appear to have been so extensively recognized as on the Continent. The principle on which its action depends, consists in causing the engine to force a quantity of atmospheric air into a reservoir with a movable cover through which the air escapes, the aperture being so regulated by an adjustable valve, that it shall only escape at a given rate. Should the speed of the engine exceed or fall short of the prescribed limit, the air is forced into the reservoir faster in the former case and slower in the latter, than it can escape through the valve; consequently the movable cover is raised or depressed, and, by means of suitable connections, partially closes or opens the throttle valve. The advantages proposed by this system are, first, greater regularity and steadiness of action than is attainable by the common governor, combined with equal delicacy; and, secondly, a more considerable range, or amount of motion available for the purpose of regulation.

Fig. 11 is an external elevation, fig. 12, a sectional elevation, (on a plane at right angles to the former), and fig. 13, a sectional plan of this apparatus.

The working parts are enclosed within a cylindrical vessel, the sole being formed of a cast-iron disc A, supported upon four small columns *a*, *a*, and the cover consisting of a light capital or cornice C of the same material; these are attached together by the four pilasters B, B, having recesses formed on their edges for the reception of cylindrical sheet-iron pannels, which thus admit of being removed at pleasure, when it is necessary to examine or repair the internal parts. In figs. 11 and 12 these pannels are shown partially removed.

Two small wrought-iron columns D, D, are also fixed to the sole plate, and serve to support a cylindrical cast-iron vessel E, the bottom plate of which is provided with two apertures guarded by the flap-valves *d*, and *g*, which open alternately for the purpose of giving admission to the air which is forced into the receiver E by the double bellows F, F′; these are respectively supplied with air from the surrounding atmosphere, by the apertures *b*, and *f*, similarly furnished with flap-valves, the former being situated in the sole-plate A, and the latter in the movable piece G; the stream of air generated in the lower bellows passing through the upper by means of an elastic leather tube *c*.* The cover of the fixed receiver E is formed of a movable cylindrical disc H, attached to the former by leather, in the manner of an ordinary bellows, and thereby admitting of being elevated or depressed, according to the degree of condensation of the air within the receiver; this is regulated by means of a small conical hole *h*, guarded by a pointed screw *i*, properly secured from turning, after being adjusted so that the air forced into the receiver when the engine is at its normal velocity, shall just have liberty to escape, and consequently hold the movable cover suspended.

Motion is communicated to this apparatus by means of two rods *l*, *l*, fixed to the movable intermediate piece G, and attached by means of the connecting rods *m*, *m*, to cranks formed on the shaft I, which is set in motion by a belt from the prime mover, working over the fast and loose pulleys J, J. A round rod K, screwed into the movable cover H, serves to convey the motion generated by the governor to the throttle valve or sluice geering, as the case may be. On this rod is fixed a ball L, which, for the sake of adapting the governor to the varying circumstances in which it may be placed, is usually made hollow, and partially filled with lead.

Fig. 16 is a representation of a mode employed by M. Molinié for rendering his governor most advantageously applicable to the regulation of the supply of water to a hydraulic motor. Besides the regular sluice-gate, he makes use of an additional valve N, to which, by means of the cord and pulleys shewn in fig. 11, he attaches the governor. The face of this valve is bent into a cylindrical form, and it is jointed by rods to a central point considerably behind the sluice face O′. By this means the strain arising from the pressure of the water against the back of the valve is counteracted, and the action of the governor rendered sufficiently delicate. Fig. 17 represents the connection of this governor with the throttle valve of a steam engine.

From the description given above, our readers will perceive that the efficient operation of this governor depends entirely on the perfection of the mechanism by which the escape of the air from the receiver E is regulated. The simple contrivance which we have detailed is altogether inadequate, as it is neither self-adjusting nor theoretically perfect in any circumstances, as will be obvious from the consideration that the volume of any fluid escaping by a given orifice depends not only on the section of that orifice, but also on the velocity of the escape; so that the higher the velocity, the aperture remaining the same, the greater will be the volume of issuing fluid.

To compensate for this defect, M. Molinié has devised an arrangement at once simple and effectual. Instead of the pointed screw *i*, he makes use of a conical pin *i*′, (see fig. 14), which is attached by nuts to the movable cover H. It is fitted to move in the interior of a brass tube h', fixed to the stationary part of the air receiver, and close at the bottom, while the top is pierced with a hole of the exact size of the thick part of the pin. The air passes by an adjustable aperture into the interior of this tube; and according as the cover H is more or less elevated or depressed, the area of the aperture of escape is proportionally increased or diminished. This ingenious contrivance not only corrects the theoretical defect above alluded to, but is attended with a great additional advantage, in the more rapid and energetic action of the governor.

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* In their more recent governors, the makers have replaced this leather tube by a small copper pipe, situated outside of the bellows, which is doubtless a great improvement.

This apparatus, although free from many of the inconveniences attending the use of Watt's governor, is yet liable to the very vital objection that its action commences still later than that of the former. The complexity of its parts, and their liability to derangement, will also prove an insuperable barrier in the way of its general adoption.

The Vane Governor.—Figs. 18, 19 and 20 represent two different modifications of the very ingenious and elegant governor invented and patented in 1841, by Mr. Benjamin Hick, jun. of Bolton. The principle of its action consists in the atmospheric resistance to rapid motion being employed to counteract the force of gravity.

The form represented in fig. 18 is that which is most applicable to the *crank-overhead* engine; and, as it illustrates the principle most clearly, we shall describe it first in order. On the crank-shaft is fixed a drum or pulley O, and underneath it, or in any convenient situation, is placed an axis carrying a small grooved pulley, to which are attached two or more fans or vanes P, P. The former communicates motion to the latter by means of an endless band or belt, which is also passed over two friction-wheels, the first of which is attached to the weighted rod r, which commands the throttle valve lever p, and the other to a gravitating weight q, suspended freely on the opposite side of the axis. The area of the vanes P, P, and the weight of the ball q, are so adjusted in relation to each other, that the latter is just sufficient to drive round the resisting vanes at a certain velocity, exactly corresponding with the normal speed of the engine. Any increase of that speed, instead of accelerating the motion of the vanes, (the atmospheric resistance being nearly uniform), tends to raise the weight, and diminish the supply of steam passing through the steam-pipe N; and, on the other hand, a relaxation of the velocity allows the weight to descend, and thereby opens the throttle valve in a corresponding proportion.

Figs. 19 and 20, are a side and end elevation of an arrangement in which this principle is carried out in a more practical and more generally applicable form. It consists of an upright spindle s, s, supported in suitable bearings in a cast-iron standard R, placed in the usual manner over the crank-shaft Q of the engine, upon which is keyed a bevil wheel, driving a pinion on the foot of the upright spindle, whereby a rapid rotatory motion is given to it. The upper part of the spindle is formed into a screw or worm, the threads of which slope at an angle of about 45°, and upon which a heavy bush or nut q, is fitted to move easily.

This bush, which is usually formed into a ball, and corresponds in its functions with the suspended weight q, in the previous example, has attached to it two or more projecting arms furnished with vanes P, P; these are so fitted upon the arms as to be capable of being set nearer to or farther from the spindle, as circumstances may require; they also admit of being turned upon the arms in an oblique direction (as shewn by the dotted lines in fig. 20), in order to diminish the atmospheric resistance. The weighted nut is connected to the throttle valve by means of a double link and swivel K, and by levers and rods M, M, n, n, in the usual manner.

From the above description it will be seen, that when the spindle s, is driven in the direction tending to raise the nut q, the latter, with its attached vanes, will be carried round with it, and at the same velocity, until and so long as the resistance of the air against the vanes corresponds with the gravitating power of the weighted nut. But when the velocity of the engine, and consequently that of the spindle s, is increased beyond that point, the atmospheric resistance against the vanes will exceed the gravitating power of the nut and its mountings, and cause them to ascend upon the screwed spindle, and thus, by means of the connecting rods and levers K, L, M, n, p, will diminish the supply of steam passing through the steam-pipe N to the engine. If, on the other hand, the velocity of the spindle is reduced below that required by the resistance of the vanes to overcome the gravitating tendency of the nut q, the latter will then descend upon the spindle, and thereby increase the passage for the supply of steam. The speed of the engine may be permanently varied at pleasure, by adjusting the vanes upon their supporting arms, so as to increase or diminish the gravitating power of the nut to the required extent.

Ingenious and beautiful as this contrivance undoubtedly is, and however admirable as a philosophical toy, it does not appear to us, either in theory or in its practical application, to possess any very decided superiority over the centrifugal governor. No doubt its capability of adjustment and unlimited range of travel are important advantages, but the substitution of the resistance of the air, which is *not uniform*, (varying as it does with the barometric state of the atmosphere and with the velocity of the vanes), for the centrifugal force of a revolving pendulum, which is absolutely uniform in all circumstances, does not appear to be a judicious arrangement. Moreover, the action of this governor must take place still later than that of the conical pendulum, and the current of air which it creates in the engine-house will be found to be an inconvenience.

The Chronometric Governor.—This is the name given to the last example in our series, by its ingenious inventor, Mr. C. W. Siemens, of Berlin; it was patented in this country in 1846. Although, in the minor details of its construction, as exhibited in our drawings, it is no doubt susceptible of material improvement, the principle of its action appears to us to be the most admirable and perfect that could be applied for the purpose of regulating the velocity of prime movers; involving as it does the happy idea of so combining the invariable motion of an *independent* pendulum with the varying speed of the engine or other motor, as to make the former correct *instantaneously* the fluctuations of the latter.

Fig. 21 is an elevation, and fig. 22 a section of this apparatus, which is set upon a bracket S, S, bolted to the wall of the engine-house, and supported by a frame-work T, T, consisting of four small columns and a circular entablature. The differential velocity between the engine and the revolving pendulum Y is obtained by means of the three bevil wheels *t*, *u*, and *v*; this last is firmly connected, by an upright spindle and grooved arm *w*, with the upper extremity of the pendulum, produced through the ball and socket joint which forms its point of suspension and revolution. The under wheel *t*, is fixed to the pulley U, which is driven by the engine with its uncertain velocity, and in the contrary direction to the motion of the wheel *v*. Both these wheels move in geer with the third bevil wheel *u*, which runs perfectly free upon its axis, and is also permitted to travel round the perpendicular socket forming the bearing of the others. It is obvious that if the wheels *t*, and *v*, revolve in contrary directions, but with equal velocities, the wheel *u*, will also revolve on its axis, but will not change its angular position; while any difference in speed between the two former will cause the wheel *u*, to follow the direction of the faster, which will at once alter the supply of steam, the arm *x*, being attached to the throttle valve contained within the steam-pipe N, by means of the lever and adjustable connecting rod *p*, and *y*. Another arm attached to the axis of the wheel *u*, on the opposite side of the perpendicular socket, is connected by means of the rod *z*, to a lever working between two adjustable stops Z, Z, which serve to confine the range of the throttle valve within convenient limits. To maintain the motion of the pendulum, a constant power is required, resembling that of the falling weight in an ordinary clock. This power is supplied by the weight *r*, which tends constantly to pull the wheel *u*, to one side; and this strain being borne equally by the wheels *t*, and *v*, causes the latter, and with it the pendulum Y, to revolve, while the former, revolving in the contrary direction, is constantly engaged to raise the weight back again into its proper position.

In practice it has been found that the power necessary for maintaining the action of the pendulum is much less than that required to effect the movement of the valve, and accordingly Mr. Siemens has adopted the principle of driving the pendulum with an excess of power which shall be neutralized by a friction apparatus when not wanted, and shall be allowed to act freely when the governor requires its assistance to move the valve. This is effected as follows:— Surrounding the grooved arm *w*, is situated a conical ring W, cast of a piece with the framing T, T, and accurately bored out. Against the interior of this "absorbing ring," a small piece of steel, accurately fitted into the open end of the grooved arm *w*, is pressed by the short end of the pendulum rod X, a spring being interposed, for the purpose of letting the pressure come on gradually. It is evident that whenever there is an excess of driving weight, which causes divergence in the arc of rotation, the surfaces of the steel rubber and of the fixed ring will be pressed together with a force exactly sufficient to produce the amount of friction necessary to counterbalance the excess; and so soon as the pendulum falls back towards a smaller arc of rotation, it will relieve the friction apparatus, and permit an increased supply of power to overcome the resistances of the valve, &c. A second spiral spring, (seen to the left in fig. 22), is laid within the grooved arm *w*, behind the point of the pendulum, for the purpose of preventing the latter from dropping into its perpendicular position, and also to facilitate its starting with the engine.

From the above description, the advantages attending the use of this governor will be manifest: the engine must always keep pace with the *independent* pendulum, which will travel only with the speed due to its perpendicular length; the amount of load upon the engine in no degree affecting the speed of the pendulum, and consequently of the engine. Further, the adjustment of the valve is effected at the very instant when the equilibrium between the load and power of the engine is disturbed; an advance of about 1-50th of a revolution of the fly-wheel is found to be sufficient to shut the valve altogether, the entire momentum of the ball Y being employed to effect that purpose. By converting the friction apparatus into a regular brake, the power of this governor may be increased almost indefinitely; in which form it may be applied directly to the shuttles of water wheels, &c. This latter arrangement will also be found very suitable for such steam engines as are furnished with variable expansion geer; it being considered most expedient to regulate the speed of such engines by increasing or diminishing the amount of expansion, rather than by throttling the steam; and the power which will generally be required for such adjustment being considerably greater than can be *directly* exerted by the ordinary governor.

PLATES CX.—CXXI.

MACHINERY FOR PREPARING AND SPINNING FLAX. BY PETER FAIRBAIRN, LEEDS.

The various combinations of machinery by which the Textile Manufactures of this country have been brought to their present high state of perfection, present a field for illustration which, however attractive, is greatly too extensive to be embraced within the limits of the present Work. These would furnish ample materials for a Work of a similar character devoted to them alone; and such a mode of treatment must be acknowledged to be more appropriate than their partial admission into a miscellaneous Series such as the present, which is addressed to readers of a somewhat different class to those who may be supposed most interested in the description of machines referred to. Some apology may, therefore, be deemed necessary in laying before our readers the present series of Plates illustrative of the preparatory processes in the Linen Manufacture. We have been induced to do so by the following considerations:— First, that while the Cotton Manufacture, and the machinery employed in it, have been made the subjects of some most valuable and widely circulated treatises, the public are left in comparative ignorance of the mechanical aids employed in that no less interesting, although, in a commercial point of view, not perhaps equally important, branch of the national industry, the manufacture of linen. Secondly, because cotton and flax machinery present, on the whole, so many points of resemblance that, to the general reader, the present series will afford an adequate idea of the kind of combinations common to both; and thirdly, because we find in flax machinery, certain distinguishing characteristics which, irrespectively of their novelty and ingenuity, will, we conceive, be especially interesting to the class of readers to whom this work is principally addressed.

GENERAL INTRODUCTION.

If we compare flax with the other materials used in the textile manufactures, such as wool and cotton, we shall find that it possesses several characteristic properties. While the latter are presented by nature in the form of insulated fibres, which simply require to be freed from their impurities previously to being spun, the former must have its filaments separated from each other by tedious and careful treatment. This circumstance has opposed so serious an obstacle to the successful introduction of machinery, as materially to have retarded the progress of the linen manufacture; and it is only within the last thirty years that the distaff and the hand-wheel have been entirely laid aside. Such rapid progress has, however, been made during that period in the improvement of flax machinery, that it has now attained to a degree of perfection little short of that by which the cotton manufacture has been so long distinguished.

To this object that eminent machinist, Mr. Fairbairn of Leeds, has contributed most materially, both by the introduction of original improvements, and by the judicious application and adaptation of the discoveries of others. From his work-shops the greater proportion of the flax machinery at present in operation throughout the world has been supplied; and accordingly we have taken his machines as the models for the present series. With one exception, (that of the hackling machine, which has not even yet been universally adopted), it will be found to comprise the entire system of machinery employed in flax factories for the preparation and spinning of the finer descriptions of yarn; the carding and spinning of the tow, which is the refuse of the hackling, being regarded as a separate subject. These drawings, with the exception of a few of the minor details, are executed to a uniform scale of one-tenth of the actual size.

Taking these machines in the order in which they are successively employed to operate upon the flax, we propose, for the sake of conciseness, after a few brief remarks upon the nature and purposes of each, to describe them by extended literal references. To render these descriptions of more value to the practical flax-spinner, we shall also append some examples of the kind of calculations necessary for the successful working of these machines.

THE DOUBLE BREAKING MACHINE.

The flax, when delivered at the mill, (having previously undergone such preparation as entirely to free it from its *boon* or woody particles, and to dissever its grosser filaments), consists of long parallel fibres of different degrees of fineness, and varying from 26 to 36 inches in length. After having been duly assorted, it is divided into ***stricks***, or small bundles, which are brought to this machine for the purpose of being cut or broken into two, three or more lengths, according to the nature of the flax, the length of the staple, and the different sorts of yarn into which it is destined to be spun. It is most generally divided into three lengths, in which case the finest and best qualities of yarn are spun from the middle portion, on account of the superior strength as well as delicacy of the fibres at that part.

Plate CXIII. Fig. 7 is a front elevation, and fig. 8 an end view of the breaking machine.

Literal References.

L, L, the frame work of the machine, consisting of two strong cast-iron cheeks, bound together by cross stays of the same material.

M, M, the strap pulleys for starting and stopping the machine; one being fast and the other loose upon the driving shaft.

N, N, the cutter or breaker, fast upon the driving shaft. It is constructed of three wrought-iron rings, firmly bolted together between two cast-iron flanges; these rings have projecting points formed at regular intervals on their peripheries, and so disposed when in combination, as to present numerous diagonal groups of cutters all round the circumference of the breaker. The cutters are of an elliptical form and project about a quarter of an inch.

O, O, a pair of grooved wheels, fixed upon a horizontal shaft at a short distance on each side of the breaker which revolves between them.

P, P, a similar pair of wheels with projections on their circumferences, fitting into the grooves on the wheels O, O, and revolving in the contrary direction.

Q, Q, a set of compound levers, rods and weights for keeping the upper grooved wheels P, P, in contact with the lower O, O. The purpose of these wheels is to bring the strick of flax under the action of the breaker, while the levers and weights Q, Q, maintain the requisite pressure upon it; the shaft carrying the upper grooved wheels being capable of moving vertically in the framing L, L.

The opposite end of the machine is provided with a set of grooved wheels, weights and levers, in all respects the same as those shown in fig. 7.

R, R, a pair of spur wheels fast upon the shafts of the upper grooved wheels, and working into

S, S, a similar pair of spur-wheels fixed to the shafts of the lower grooved wheels O, O, which is driven by a train of wheels at the opposite side of the machine. The wheels R, R, S, S, are made with teeth of a somewhat greater length than usual, to allow the flax to pass between the grooved wheels without disengaging the connection.

T, T, T, a train of spur-wheels and pinions by which the motion of the driving shaft is communicated, with a greatly reduced velocity, to the grooved wheels O, O, P, P.

Action of the Machine.—The strick or small bundle of flax is held in both hands, and laid transversely against the grooved wheels O, O, P, P, which rotate slowly in the direction of the breaker. The latter, which makes about 300 revolutions per minute, tears or breaks it across by separating but not cutting the fibres. Two hands may be employed at the same time at this machine, one at each side.

The coarser having been thus divided from the finer portions of the flax, each sort is collected into a separate heap, and it is then ready to undergo the next process, namely, the hackling.

HACKLING, AND HACKLING MACHINERY.

This is an operation of the utmost importance in the preparation of flax, as its successful accomplishment affects not only the quality but the quantity, or *yield*, of the yarn into which it is ultimately converted. The object proposed by it is threefold: First, the parting of the filaments into their finest fibrils; secondly, the separation and removal, with the least possible injury to the real fibres, of the *tow*, or short fibres which adhere to the former and bind them together; and thirdly, the equable and parallel arrangement of the long filaments. To accomplish these objects, the flax is drawn successively over two or more surfaces, thickly studded with sharp steel pins, or *hackle teeth;* the thickness of these, and their degree of approximation varying with the progressive stages of the operation.

Simple as this process appears, the most careful and dextrous manipulation is required for its efficient and economical accomplishment; and it may be doubted whether any combination of machinery can produce as good work with as little loss as hand labour. In hackling by the hand, the operative wraps one end of the strick of flax firmly over his right hand, throws it upon the points of the coarse hackle, and draws it towards him, while he holds the left hand upon the other side of the hackle, in order to spread the flax, and to prevent it from sinking too deeply among the teeth. The other end of the strick is treated in the same manner; and the operation is repeated, first upon the coarse and then upon the fine hackles, until the fibres have been reduced to a sufficient degree of fineness, and, by careful handling, little more tow can be formed. To facilitate subsequent operations, the ends of the strick of flax should assume, at this stage of the process, an elliptical form when laid on a flat board. For this purpose, the hackler wraps the end of the dressed strick round a pin shaped like a knife, and breaks or pulls away such of the fibres as, by their length, would tend to give it a pointed instead of an elliptical shape.

As we have before had occasion to remark, the various mechanical contrivances which have hitherto been employed in the hackling of flax have not been found so successful as to have entirely superseded manual labour. In hackling by the hand, the workman feels at once the degree of resistance, and can accommodate the traction to it, or throw the flax more or less deeply among the teeth according to circumstances, as well as draw it through them with suitable force and velocity. It will be readily understood that these circumstances present a serious, though not, perhaps, insurmountable obstacle to the successful application of machinery. Many ingenious and beautiful machines have, however, been employed for this purpose, and the performance of some of them has been found highly satisfactory. We shall attempt to convey an idea, without the aid of a drawing, of one of the best of these, the cylinder hackling machine, as improved by Mr. Bush of Leeds.

This machine consists of two horizontal cylinders, about 3 feet in diameter, on the peripheries of which are fixed hackles; the strength and number of these being varied according to the different stages of the operation. These cylinders are fixed upon shafts which are made to revolve with a considerable velocity in directions contrary to each other, both cylinders moving towards the centre of the machine. At each end of these cylinders is placed a large spur wheel, geering with a central wheel situated between the two hackle-cylinders; this last is called the *transfer wheel.* All these wheels are fitted with a number of segments fixed to their inner surfaces, and forming a species of drums, of a diameter considerably greater than that of the hackle-cylinders. Each of these drums contain a series of recesses into which are fitted the clamps or *strick-holders* by which the flax is secured. The drums and hackle-cylinders are set with a considerable degree of eccentricity in relation to each other, so that the rim of the external drum may be farther from the hackles at the back part of the machine, where the operation commences, than at the part when the strick is delivered to the transferring mechanism; this arrangement is requisite for the purpose of subjecting the end of the strick to the action of the hackles for a longer period than the middle, where the fibres are naturally much stronger and finer, and do not contain so much tow. The operation of this machine is as follows: the stricks having been firmly secured to their holders, the latter are inserted into the recesses of the first external drums; these rotate with a slow motion towards the centre of the machine, and the hackle-cylinders, revolving within them in the same direction but with a much greater velocity, dress the *under* sides of the stricks. On reaching the transfer wheels, the strick-holders are, by an ingenious combination of slides and weighted levers, disengaged from the back cylinder, and pushed into the recesses on the transfer wheels; by these they are inverted and conveyed to the front cylinder and drums, which complete the process by hackling the *upper* side of the flax.

On being delivered at the front of the machine, which we have supposed to be the first, or *roughing* hackle, the stricks are conveyed by the attendant to the finishing machine, which is in all respects identical with that described, but is furnished with finer hackles. After passing through this, the strick-holders are unscrewed, and the undressed ends drawn through them; they are again secured and returned to the roughing machine, where, by a repetition of the process already described, the opposite ends are finished.

The tow is cleared from the hackle-cylinders by means of revolving brush-cylinders, situated at each end of the machine, and underneath the centres of the former. From the brush-cylinders it is again transferred by doffing cylinders covered with card filleting; these latter are cleared by means of doffing knives, worked by eccentrics on the end of the brush-cylinders, the tow being delivered in broad sheets on the floor of the factory, or into proper receptacles. The carding and spinning of the tow is performed by machinery similar to that employed in the cotton manufacture, though of considerably greater strength.

THE SPREADING MACHINE.

The next process undergone by the flax is its conversion into an endless band of parallel and rectilinear filaments, called a *sliver*, being the foundation of the future yarn. This is accomplished by the machine now under notice, which is fully delineated in Plates CX., CXI.

Fig. 1 is a front elevation of the spreading machine; fig. 2, a side elevation; fig. 3, a general plan, and figs. 4, 5 and 6, detached views, on an enlarged scale, of the spirals and fallers.

Literal References.

A, A, A, the cast-iron frame work of the machine.
B, B, the driving shaft, fitted with a fast and loose pulley.
C, C, the feeding or *spreading* table; it is divided into two compartments, the one being considerably longer than the other, for the convenience of the attendants who spread the flax.
D, D, D, rollers situated at each end of the feeding table; over these pass four endless leather straps upon which the stricks of flax are spread.
E, E, a polished iron plate, upon which are fixed the guides which serve to conduct the flax to the back or detaining rollers.
F, a cylindrical tin can, placed in front of the machine to receive the sliver.
a, *a*, *a*, the front or lower *drawing roller*.
b, *b*, *b*, the top drawing or *pressing* rollers, made either of wood or iron, and covered with leather.
c, *c*, *c*, the back or *detaining* rollers.
G, G, two weighted levers for imparting the requisite pressure to the top drawing rollers.
H, H, two weighted levers bearing in a similar manner upon the top detaining roller.
d, *d*, the *fallers* or *gill bars*, forming a sheet of advancing hackles between the detaining and drawing rollers; these are for the purpose of producing great regularity in the draught, and a perfectly parallel distribution of the fibres.
e, *e*, *e*, the *rubbers* for clearing the top drawing rollers from adhering fibres.
f, *f*, *f*, brass guides for conducting the sliver to the drawing rollers.
g, *g*, the *sliver-plate*, formed with bevilled openings for the sliver to pass through towards
h, *h*, the *calender rollers*, by which the four slivers are compressed into one, and delivered in the form of a riband into the can F.
K, the calender roller shaft.
i, *i*, cast-iron hangers for transmitting the pressure of the weighted lever G to the top drawing rollers.
k, *k*, are similar hangers attached to the lever H.
l, *l*, the spirals or screws; into the spaces between the threads of which, the ends of the fallers are inserted.
m, *m*, two pairs of small bevil wheels by which the lower spirals are driven from the *back shaft*.
n, *n*, two small spur wheels communicating motion from the lower to the upper spirals.
o, *o*, the tappets or *cams* by which the fallers are elevated in succession from the lower to the upper spirals, and *vice versa*.
p, *p*, small weighted levers for guiding the fallers between the threads of the spirals *
q, a small endless screw cut upon the extremity of the axis of the lower drawing rollers *a*, *a*; it works into
r, a worm-wheel, on the axis of which is another endless screw, driving a similar wheel, called the *bell-wheel*; at every revolution of this last wheel, a pin fixed into its rim acts upon a spring L, to the end of which a bell is attached. Thus the ringing of this bell serves to register the length of the sliver delivered into the can.

The following is the detail of the wheel work in this machine:—On the driving shaft B is fixed the spur pinion *s*, working into the wheel *t*, on the lower drawing roller shaft; to this latter axis is affixed the wheel *u*, whose motion is communicated by the movable intermediates *v*, and *w*, to the change pinion *x*, on the back shaft; the relative diameters of these wheels regulating the amount of *draught*, or the degree of extension which the flax sustains in passing between the detaining and drawing rollers. The opposite end of the back shaft carries the pinion *y*, working into the stud-wheel *y′*, having on its boss the pinion *z*, which, by means of a movable intermediate *z′*, drives the wheel 1, on the axis of the detaining roller. This train of wheel work is calculated to produce a nearly uniform surface speed of the rollers and the sheet of hackles.

A slow motion is communicated to the *sheet roller* D, over which the feeding bands pass, by means of the spur wheel 3, working through an intermediate 2, into the pinion *z*.

A uniform velocity is imparted to the lower drawing and calender roller shafts *a*, and K, by a pinion 4, on the extremity of the former, working through an intermediate wheel 5, into a similar pinion 6, on the end of the latter.

And lastly, a revolving brush situated under the lower range of fallers, for the purpose of clearing away the dust, is driven by the stud-wheel 7, geering with the pinion 4, and having on its boss a small pinion working into the wheel 8, on the end of the brush-shaft.

Mode of Calculating the Draught.—By an attentive consideration of what has been said above regarding the draught wheels between the back and front rollers, it will be seen that the differential velocity of their peripheries is produced by the following wheels, which we have here noted in their successive orders:

u, a driving wheel on the end of the front roller.
x, a driven do. on the end of the back-shaft.
y, a driving do. on the opposite end of do.
y′, a driven do. working on a stud.
z, a driving do. on the boss of the stud-wheel *y*.
1, a driven do. on the end of the detaining roller.

The relative diameters of these wheels, together with those of the drawing and detaining rollers themselves, regulate the draught; the remaining wheels of the train being merely introduced for the purpose of conveying motion from one point to another.

In calculating the draught, these wheels and rollers are to be set over against each other in the following order, observing that the back rollers always rank with the drivers, and the front with the driven.

DRIVERS.	WORKING INTO.	DRIVEN.
u, No. of teeth, say 39	....	*x*, No. of teeth, say 25
y, ... 14	....	*y′*, ... 38
z, ... 10	....	1, ... 33
Diam. of back roller, 3 in.		Diam of front roller, 3⅝ in.

$$\frac{25 \times 38 \times 33 \times 3\frac{5}{8}}{39 \times 14 \times 10 \times 3} = \frac{113643\cdot75}{16380} = 6\cdot93 \text{ the draught}$$

of spreading machine.

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* We shall have occasion, in describing the machine which comes next under notice, to give a full explanation of the mode in which the gill-bars are worked; at present, accordingly, we merely enumerate the parts of the mechanism.

Mode of Calculating the Length of Sliver.—To calculate the length of sliver delivered into the can F between each revolution of the bell-wheel, or ringing of the bell; Multiply the circumference of the drawing roller *a*, in inches, by the number of teeth in the worm-wheel working into the endless screw *q;* the product multiplied by the number of teeth in the bell-wheel will give the number of inches delivered during one revolution; which divided by 36 gives the number of yards in each can.

Example.—Suppose the circumference of the drawing roller 11 inches, and the number of teeth in each worm-wheel 60, the following is the result:

$$\frac{11 \times 60 \times 60}{36} = \frac{39600}{36} = 1100 \text{ yards, the length per}$$

can for each revolution of the bell-wheel.

Action of the Machine.—The flax is placed in the sheet-iron guides behind the detaining rollers, and along the endless bands or feed sheets, by laying down one handful after another, so that the points of the second strick reach to about the middle of the first, and thus preserve a uniformity of thickness in the feeding. By the motion of the machine it is introduced between the back rollers *c*, *c*, and carried forward by the sheet of hackles *d*, *d*, towards the front or drawing rollers *a*, *a*, *b*, *b*, which, revolving at a velocity considerably greater than the former, lengthen or draw it out to a proportional extent; the hackles at the same time combing, separating and straightening the fibres. The slivers from the four drawing rollers are then passed through the bevil slits in the sliver-plate *g*, *g*, and united into one by the calender rollers *h*, *h*, by which they are subjected to a gentle pressure and delivered into the can F. This union of the slivers is necessary in order that the varying thicknesses of each may be compensated, and perfect uniformity attained. When the can has received its destined supply, the ringing of the bell warns the attendant to break the flax, remove the can, and substitute another.

THE DRAWING FRAME.

The next process in the preparation of flax consists in causing it to pass twice in succession through the drawing machine, for the purpose of still further increasing the fineness and uniformity of the sliver.

This machine, which is represented in figs. 1, 2 and 3, Plates CXII., CXIII., is in principle identical with, and in the details of its construction, very similar to, the spreading machine already described. It contains, as will be seen by the drawings, two sets of fallers and rollers, and the place of the feeding table and guides is supplied by a bent plate of polished sheet-iron C, extending across the entire breadth of the machine, over which the slivers glide in passing from the cans to the detaining rollers. These latter differ slightly from those used for the same purpose in the spreading machine; here they are three in number, coupled together by small pinions, 1, 2, and 3, and disposed in a triangular form, the sliver being made to pass under the first, over the second, and under the third. With these exceptions, there is no essential difference between the present machine and that last described, and as we have been careful to designate the same or analogous parts in both by the same letters, it will be unnecessary to repeat the description.

The Spiral or Screw Gill.—Figs. 4, 5 and 6, Plate CXIII., give a representation of a very interesting and important piece of mechanism which enters largely into the construction of modern flax machinery. This ingenious contrivance, which is called the *spiral* or *screw gill geering*, was patented by Messrs. Westley and Lawson in 1833, and has, in a great measure, superseded all former modes of effecting the same object, which is the combing and separating the fibres of the flax, in order to facilitate the drawing and to give uniformity to the sliver.

The fallers or hackle bars *d*, *d*, (see fig. 5), are supported at both extremities by the horizontal steel guide rails *k*, *k*, screwed to the insides of the sockets in which the spirals or screws *l*, *l*, work; these sockets being bored in projecting parts cast upon the stands D, D. The lower screw is driven from the back shaft by a bevil wheel and pinion *m*, *m'*, and a small spur-wheel *n*, on the back of the bevil pinion, works into a corresponding wheel on the top screw, driving both screws at equal velocities but in contrary directions. In the sides of the sockets in which the screws revolve, openings are formed, parallel with their axis, and coinciding with the surfaces of the guide-rails *k*, *k;* through these openings the ends of the gill-bars, (which are steeled and bevilled to compensate for the angle of the threads), are inserted into the helical grooves of the screws. Thus the rotatory motions given to the spirals cause the fallers to be driven along the guide-rails in a vertical position, and with a uniform simultaneous movement; the top sheet in the direction of the drawing rollers, and the lower towards the detaining rollers.

On reaching the front part of the machine, close behind the drawing rollers, the fallers are depressed and put out of operation by means of the rotatory cams *o*, *o*, fixed to the ends of the top screws; being guided vertically downwards between the ends of the upper guide-rails and the weighted levers *p*, *p*. They are thus engaged in the threads of the lower screws, which carry them to the opposite end, where similar cams *o*, *o*, and weighted levers *p*, *p*, raise them successively into their original position on the upper guide-rail, where they are traversed forward as at first. It is usual to make the lower spiral with its threads considerably wider than those of the upper, which arrangement diminishes the total number of fallers requisite for the due performance of the work.

The Chain or Sheet Gill.—Having described the modern method of traversing the gill-bars, it may not be uninteresting to our readers to be furnished with an idea of the mechanism employed before its introduction for that purpose, especially as the latter is still in very general use, and possesses at least one important advantage over the former.

In the old system, which we have illustrated by the accompanying wood-cut, the sheet of hackles is carried forward by being connected with an endless chain of alternate single and double brass links, working over the two parallel axes *a*, and *b*, which carry, at each end, four small cams or teeth, adapted to work between the bottoms of the double links. Motion is communicated to the shaft *b*, by a train of wheels from the drawing roller, and a corresponding motion is imparted to the shaft *a*, by a train coupling it with *b*. Each link of the endless chain carries a species of fork *d*, *e*, *f*, close at the outer extremity, the space between the prongs being designed as a guide for the ends of the fallers. These are restricted to their destined course by lateral slides of a somewhat irregular form, which are bolted firmly to the carriage plates by the pieces *c*, *c*, *c*, cast upon them; between these slides the round ends *j*, *j*, fig. 2, of the fallers are inserted, after having been passed through the forks *d*, *e*, *f*; a plate *g*, is fastened to the slides for the purpose of giving rigidity. By the arrangement above described, it will be seen that as the endless chain moves along, the fallers *i*, *i*, will move with it, and will, at the same time, travel in a line determined by the form given to the lateral slides. This is such, that the points of the hackles *h*, *h*, do not extend beyond the ends of the forked guides until they arrive at that part where they should come into contact with the flax.

Fig. 1.

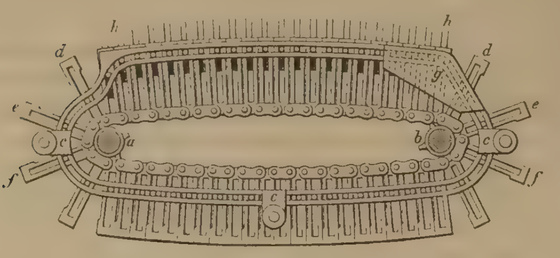
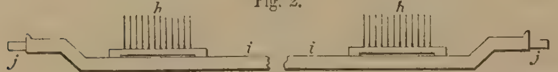


Fig. 2.



Comparison of the Two Methods.—In estimating the comparative merits of the screw and sheet gills, it will be obvious that, besides the much greater complexity of the latter and its consequent liability to derangement, the former possesses the very great advantage of being made to act perpendicularly upon the sliver, immediately after it leaves the detaining rollers; and of continuing its action throughout the entire space between the detaining and drawing rollers, causing the hackles to drop vertically downwards close to the latter. This, it is scarcely necessary to point out, cannot be effected by the sheet gill, and a certain amount of irregularity in the drawing of the sliver is the consequence. On the other hand, it possesses an advantage over the screw gill in the facility which it affords for allowing the hackles to be cleaned by a revolving brush during their passage along the under side of the slides; this can only be very imperfectly accomplished in the screw gill, on account of the points of the hackles being always turned upwards.

THE SELF-REGULATING SPIRAL ROVING MACHINE.

The conversion of the slivers of flax, as of all other materials used in the textile manufactures, into yarn, is effected by the combined operations of drawing and twisting. In the later stages of the process, these operations must be carried on simultaneously, since it is obvious that the elongated slivers of parallel filaments could not, without breaking asunder, bear extension to the requisite degree of tenuity without being slightly entwined together. The amount of twist given to them in the first instance must not exceed that which is necessary to give cohesion, otherwise their subsequent attenuation into a regular thread will be obstructed. The final drawing of the *slivers*, their conversion into a species of coarse loose yarn, and the winding of the latter upon bobbins for greater convenience in the subsequent process, are effected by the machine now before us.

In the present highly improved form of this important and complicated machine, it may be regarded as a striking triumph of mechanical ingenuity. By the separate application of the patent spiral driving pinions[*] to the spindles and bobbins; by the introduction of the screw gill machinery described above; and by the adoption of numerous movements employed in the cotton manufacture, the roving frame has been brought by Mr. Fairbairn to a very high state of perfection. Our limits will not admit of our giving an adequately detailed account of this machine, and of the admirable manner in which it performs its numerous functions; we shall, however, endeavour to convey to our practical readers such an idea of it as will enable them, with a little study, to master the details.

Enumeration of the Plates.

Plates CXIV., CXV., present a general elevation of the entire machine as seen in front; one of the plates which protect the spiral pinions being removed in order to show the mode of giving motion to the spindles, flyers and bobbins.

Plates CXVI., CXVII. A corresponding general elevation of the back part of the roving frame, exhibiting the cone and differential movements, and by the removal of one of the covering plates, exposing a part of the back shaft and geering for working the fallers.

Plate CXVIII., Fig. 3 is an end elevation, showing the principal geering employed in this machine; on each wheel the direction of its motion is denoted by an arrow.

Fig. 5 is an elevation, partially sectioned, of a portion of the spindle-rail or beam, and fig. 6 is a plan corresponding.

Fig. 7 is an elevation, also partially sectioned, of part of the bobbin-lifter with its attached geering, and fig. 8 is a plan of the same.

Plate CXIX., Fig. 4 is a transverse section of the machine, exhibiting some of its internal arrangements, and showing the course of the sliver from the cans to the bobbins.

Fig. 9 is an elevation of part of one of the stands, showing the slides, springs and weighted levers used for defining the course of the fallers.

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[*] The principle and mode of action of these pinions will be found fully discussed at page 79 of our Treatise on Geering.

Fig. 10. An elevation and plan of the contrivance for transmitting motion to the axis of the spirals on the bobbin-rail.

Fig. 11. A cross section of a part of the machine, showing the apparatus for maintaining constant tension upon the strap driving the cone pulley of the traverse and equational bobbin motions.

Literal References.

A, A, the driving shaft, situated towards the back of the machine, and extending throughout its entire length.

B, B, a shaft parallel and near to the driving shaft, extending from the centre to one end of the machine. It carries a spur wheel at each extremity, one of which is commanded by the equational bobbin motion, while the other, by means of a peculiar arrangement of geering, to be hereafter described, transmits the motion to the spiral-shaft on the bobbin-rail.

C, C, the mangle-pinion shaft, worked by a train of bevil wheels from the cone-shaft, and, through the mangle wheel situated at one end of the machine, close to the driving pulleys, working

D, D, the mangle-wheel shaft, extending the whole length of the machine, and carrying pinions working into racks 8, 8, 8, attached to vertical slides; these slides are furnished with projecting arms fixed to the bobbin-rail, which is thus traversed up and down by the mangle wheel; causing the flyers to wind the roving between the flanges of the bobbins with all the regularity of a screw. Counter-balance weights 1, 1, 1, attached to the bobbin lifter by means of chains passing over pulleys, serve to relieve the racks from all unnecessary strain.

E, a short shaft situated at the back of the machine, and driven by a train of spur wheels from the driving shaft at the same velocity as the latter. Upon this shaft is suspended a species of frame, fitted to slide longitudinally upon it, and carrying two pulleys and a weight at the extremity; the first pulley being adapted to rotate with the shaft E, by means of a long slot and sunk feather, and the other being merely a conical friction roller, for the purpose of maintaining a constant tension upon the strap driving the cone F (see fig. 11). The frame, with its appendages, is traversed along the shaft by means of a weight 2, situated at the opposite end of the machine, and attached by a chain and adjustable rod to the frame, and to a rack working in a slide fixed to the back of the roller-beam (fig. 4). This rack is serrated on both edges, the teeth of the upper alternating with those on the lower edge, and the pauls are alternately disengaged at every revolution of the mangle wheel in such a manner as to allow the drag-weight to advance the rack and pulley-frame by half the distance between two contiguous teeth. The mechanism by which the pauls are disengaged is as follows:—At the back of the rack-slide a short rectangular bar is fitted to slide vertically, having two projecting pins acting upon the points of the clicks (see fig. 4). This bar is worked by the end of a lever inserted through its lower extremity, and having its fulcrum under the roller-beam; to the other end is attached the vertical rod 5, carrying two adjustable catches 4, 4, which, at every alternate movement of the mangle wheel, are struck by an arm 6, extending from the rack 7. Thus one of the pauls is constantly in geer to prevent the rack and attached pulley-frame from being drawn beyond the prescribed limits for each stroke. The lower paul is kept pressed against the rack by a counter-weight 3, while the upper one merely rests on it by its own gravity. The pitch of the teeth on the rack must be varied according to the degree of fineness of the roving.

F, the cone, driven by a strap belt from the pulley on the shaft E, and communicating a gradually retarded motion, at once to the bobbins themselves, and to the traverse of the bobbin-rail; the velocity of the spindles and flyers remaining constant. This is necessary in order to compensate for the continually increasing diameter of the bobbins, as the roving is wound upon them. The cone is set at a slight inclination, in order to allow the belt to act upon a greater part of its periphery towards the apex than towards the base, where, on account of the increased diameter, this precaution is less necessary to ensure the rotation of its shaft. A lever-handle (seen in the end elevation, fig. 3), is attached to the carriage of the cone shaft, for the purpose of raising its outer extremity previously to winding back the pulley-frame.

G, H, two short shafts situated towards the centre of the machine, and carrying geering to be hereafter specified, for transmitting the motion of the cone-shaft F to the traverse and equational bobbin motions respectively.

I, a hollow boss fitted to rotate upon the driving shaft A, with a motion independent of that of the latter, and carrying at one end a spur pinion working into a wheel on the shaft B, and at the other a bevil wheel, being part of the equational bobbin motion.

K is a similar bevil wheel fast upon the driving shaft A.

L, L, the back shaft, traversing the entire length of the machine, and carrying the bevil wheels for working the mechanism of the fallers.

M, M, a longitudinal shaft working in bearings on the spindle rail, and carrying the spiral pinions for conveying motion to the spindles and flyers (see figs. 4, 5, and 6). The spindles are disposed in two rows, so that each spindle in the back range stands opposite to the interval between two in the front range. The object of this distribution is economy of space, as the machines would require to be greatly longer if the spindles stood all in one line. The shaft M is situated between the two rows, and drives both rows in the same direction.

N, N, a longitudinal shaft working in bearings on the bobbin-rail, and carrying the spiral pinions for working the bobbins (see figs. 4, 7, and 8). The spindles pass through brass sockets fixed to the bobbin-rail to hold them steady as the latter traverses up and down. These sockets serve also as pivots for the spiral pinions to revolve upon; it being understood that the motion of the bobbin spirals is totally independent of that of the spindles. A small flange on the top of each spiral carries two projecting pins fitting into corresponding holes in the bottom of the bobbins, and causing both to revolve together.

O, P, the back detaining rollers having the iron pressing rollers between them, which are cut into short lengths, and are carried round by the friction caused by their own gravity.

Q, Q, the axes of the pressing or top drawing rollers marked 10, which are usually made of wood, and are pressed against the lower drawing rollers 12, 12, by hangers 11, 11, resting in necks cut in the axes, and attached to weighted levers 9, 9, passing under the roller-beam.

R, R, a slender rod extending the entire length of the machine, for conducting the slivers to the detaining rollers. They pass under this rod, and slide over a polished sheet-iron plate covering the back shaft and bevil geering for driving the gill-screws.

S, S, a rod by which the attendant is enabled at any part of the machine to stop or set it in motion.

T, T, are a set of friction pulleys placed upon a rod surmounting the machine, for the purpose of guiding the slivers as they are drawn out of the cans by the action of the machine.

U, V, the fast and loose pulleys on the end of the driving shaft for starting and stopping the machine.

W, W, W, is a train of spur-wheels conveying the motion of the driving shaft A to

X, a spur pinion on the end of the shaft M, which drives the spindles and flyers with a uniform motion.

Y, Z, the draft geering between the drawing and detaining rollers, the particulars of which will be given below.

a, *b*, *c*, and *d*, are spur and bevil wheels and pinions, the combination of which forms the differential motion for driving the bobbins. The principle of this motion and the details of its construction are fully treated of (though in reference to a totally different object), at page 146 of our Treatise on Geering.* Referring, therefore, to that article for the above particulars, we add the following remarks in explanation of its present application. Supposing that the large spur wheel *a*, which, through the pinion *b*, receives its motion from the cone, were driven at the same velocity as the driving shaft A, then it is obvious that no motion whatever would be imparted to the bobbins. On the other hand, if the wheel *a*, were held absolutely immovable, the bevil K, which is fixed upon the driving shaft, would convey, through the pinions *c*, *c*, a motion equal to its own (though in the contrary direction,) to the boss I and attached geering; consequently, in the case we have last supposed, the motion communicated to the bobbins would be uniform. Hence, by combining the two extreme cases, and supposing the wheel *a*, to be driven in the direction of the driving shaft, but with a slower velocity, it will be understood that the boss I will be made to revolve at a speed which, if added to that of the wheel *a*, will be exactly equal to that of the driving shaft A.

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* The merit of the first application of this ingenious and beautiful contrivance to spinning machinery is due to Henry Houldsworth, Jun., Esq. of Manchester, who introduced it into the cotton bobbin and fly frame. Previously to its application, the motion of the bobbins was regulated by drag-weights, which required to be shifted by the attendant, as will be explained in treating of the spinning machine.

Thus, when the driving strap is at the apex or starting point of the cone F, the wheel *a*, is at its maximum velocity, and the boss I with the train of wheels to the bobbins at their minimum, causing the flyers, which revolve at a considerably greater *uniform* speed, to coil the given quantity of rove upon the bobbins; then as the strap advances towards the base of the cone, (every point in this advance being simultaneous with the commencement of a fresh layer of roving), the speed of the wheel *a*, is diminished, causing that of the boss I to *increase* in the same ratio, and thus approximating the speed of the bobbins to that of the flyers, at every alternate motion of the traverse. In this way the irregularity due to the varying diameter of the bobbins is compensated, and a uniform very slight tension maintained upon the slivers, between the flyers and the drawing rollers.

e, *e*, *e*, a train of bevil wheels and pinions for conveying the motion of the short shaft G to the mangle-pinion shaft C.

f, *g*, *g*, a train of spur wheels and pinions, (including change pinions), for conveying the motion of the cone F, at once to the traverse and equational motions. It is obvious that to preserve the regularity of the winding, the speed of the traverse or *copping* motion, as well as that of the bobbins themselves, must be progressively retarded.

h, *h*, *h*, a train of spur-wheels for conveying the differential motion to the bobbin shaft N. The pinion *d*, on the boss I works into a wheel fixed to the end of the shaft B; this shaft has another spur wheel *h*, (see figs. 3 and 10), upon its opposite extremity, which geers with *h*, an intermediate wheel suspended in a joint formed by the meeting of two pairs of arms; one of which has its centre of motion on the shaft B, and the other upon the shaft N. Thus, when the latter ascends and descends in obedience to the traverse motion, the arms move radially round their respective centres, and consequently the suspended wheel *h*, is kept constantly in geer both with the wheel on the end of the shaft B, and with the pinion on the shaft N. This will be clearly understood by observing the dotted lines in fig 10, which denote the different positions of the bobbin-lifter, and the corresponding positions of the arms and intermediate wheel.

i, *k*, *l*, *m*, a train of spur wheels for conveying the motion of the driving shaft to the shaft E working the cone motion, (see fig. 3).

n, is a spur wheel on the end of the drawing roller, also working into the movable intermediate *k*, which thus commands the drawing, the bobbin, and traverse motions. The train *i*, *k*, *n*, is called the *twist geering*, and its object is to vary the speed of the front roller while the speed of the spindles remains the same, and thus to put more or less twist into the rove, as may be required.

o, *o*, *p*, a train of wheels between the drawing rollers and the back-shaft—*p*, being a change pinion; these, together with the train Y, Z, at the opposite end of the machine, constitute the draught geering.

r, *r*, small pinions connecting the detaining rollers together.

t, *u*, a handle and small bevil wheels working a barrel round which is coiled a chain attached to the pulley-frame for winding the rack, &c. towards the apex of the cone F.

v, *v*, *w*, *w*, *y*, *y*, the fallers and geering for working them, as minutely detailed in a preceding description.

x, *x*, *x*, the flyers fixed upon the top of the spindles for twisting, guiding and winding the rove upon the bobbins.

Calculation of the Draught.—The arrangement of the draught geering in this machine is precisely similar to that in the Spreading and Drawing Machines, and the mode in which the calculation is made need not be repeated. With the wheels employed in our Plates and the diameters of the drawing and detaining rollers being respectively 2 inches, the result of the calculation will be as follows:

$$\frac{38 \times 60 \times 52 \times 2}{58 \times 28 \times 16 \times 2} = \frac{237120}{51968} = 4{\cdot}56,$$

the draught of roving.

Calculation of the Twist.—The amount of twist given to the roving depends upon the relative velocities of the spindles and of the surface of the drawing rollers. This may be varied at pleasure by putting a larger or smaller wheel *i*, upon the end of the driving shaft. The elements which enter into the calculation of the twist are these:

W, a driving wheel on the driving shaft A.
X, a driven pinion on the spindle spiral wheel shaft M.
The spiral driving wheels on the shaft M.
The spiral driven pinions on the spindles.
i, a driving wheel on the driving shaft A.
n, a driven wheel on the end of front roller.

These are classified thus:

DRIVERS.	WORKING INTO.	DRIVEN.
W, No. of teeth, say 60	....	X, No. of teeth, say 32
Spiral wheel,....... 18	....	Spiral pinion, ... 12
i,...................... 46	....	*n*,....... 96
Circumference of front roller 6·28.		

$$\frac{60 \times 18 \times 46}{32 \times 12 \times 96} = \frac{49680}{36864} = 1{\cdot}34,$$

the twist for each revolution of the drawing roller. This divided by the number of inches in the circumference of the drawing roller will give the twist per inch: $\frac{1{\cdot}34}{6{\cdot}28} = {\cdot}21$.

To describe the operation of this machine would be merely to recapitulate most of the details already given; we shall, therefore, pass on to the final process, the Wet Spinning of flax.

THE WET SPINNING FRAME.

With the exception of the hot-water trough and its adjuncts, this machine bears a close resemblance to the *Throstle Frame* of the cotton manufacture, and like it, is employed for the completion of the yarn after, by being repeatedly subjected to the processes of drawing and roving, until it has attained the requisite degree of attenuation.

Although the principle of its operation be, for the most part, the same as that of the roving machine, it is much less complicated than the latter, inasmuch as it dispenses with the gill apparatus, (which is, of course, only applicable to parallel slivers), and with the equational bobbin motion, which, in the present instance, is rendered unnecessary by the circumstance of the yarn itself having attained a sufficient degree of cohesive force to enable it, with the aid of a simple contrivance, to regulate the drag upon the bobbins.

We are indebted to Messrs. Marshall of Leeds for the very important improvement in the spinning of flax, introduced by the use of the hot-water trough, through which the rovings are caused to pass in the act of spinning. By this means, the glutinous matter adhering to the fibres is thoroughly dissolved, and a much finer, smoother and more uniform thread than could otherwise have been possible, is produced. The improvement in the method of driving the spindles, by which one tape is made to communicate motion to four spindles, is also due to Messrs. Marshall. Another improvement deserving special notice, both for its ingenuity and efficiency, consists in the method of working the traverse motion, as figured in our Plates. To Mr. Fairbairn we are indebted for this motion, which has superseded the old imperfect system of working the traverse by means of a heart-wheel, levers and chains.

Plates CXX., CXXI., give representations of the improved Flax Spinning Frame.

Fig. 1 is a front elevation, broken in the middle, for the purpose of exposing part of the geering for working the traverse motion.

Fig. 2 is an elevation of the *geering* end of the machine; and fig. 3 is a transverse section of the entire machine.

Literal References.

A, A, the frame ends or standards, of cast-iron.

B, B, the middle support.

C, C, longitudinal beams of cast-iron, on which are supported

a, a, the stands or framing of the rollers. The bottom or drawing roller journals are fixed, while the top or detaining roller journals slide on the upper part of the stands, and are regulated by screws, (see dotted lines in fig. 3), so as to adapt the *reach,* or distance between the drawing and detaining rollers to the various lengths of fibres. This distance should always be a little more than the average length of the filaments.

b, b, the bottom or drawing rollers, usually called the *front* rollers.

c, c, the top or detaining rollers, usually called the *back* rollers. Both front and back rollers are made of brass cast upon a wrought-iron spindle and fluted.

d, d, the *saddles* for retaining the pressing rollers in their proper places. The bushes or bearings of the top pressing rollers are made to slide upon projecting arms in order to suit the various lengths of reach. The top pressing rollers are generally made of brass and the bottom of box-wood, and both are fluted.

e, e, are bolts fitted with adjusting thumb-screws for attaching the saddles *d, d,* to

f, f, g, g, levers with weights for giving the requisite pressure to the pressing rollers.

h, h, cranked axles extending the entire length of the machine, for relieving the pressing rollers from the strain of the weighted levers when the machine is at rest.

i, i, ratchet-wheels fixed upon the end of the cranked axles for maintaining them in the position in which they may be placed.

D, D, the wooden troughs surmounting the machine, and through which the rovings pass before reaching the detaining rollers. These troughs are supplied with hot water, and kept at a high temperature by steam from a boiler.

E, the *creel* in which the roving bobbins are placed vertically, in alternating rows.

F, a wooden rail surmounting the creel, to which are attached

j, j, the slender sheet-iron supports for the top of the roving bobbin spindles, the lower ends revolving in footsteps on the top of the trough, (see fig. 3).

k, k, the bobbins, as filled with loose yarn by the roving machine.

l, l, l, l, longitudinal brass rods for conducting the rovings into and through the troughs D, D.

m, m, is a flat brass rod placed immediately above the detaining rollers, and extending the entire length of the machine; opposite to each boss of the detaining rollers an indentation is cut in the rod *m,* for the purpose of guiding the rovings. A small endless screw is cut on the end of the detaining roller shaft, and geers with a worm-wheel *p,* (see fig. 1), which works on a stud fixed to the beam C, and has a small *heart-wheel* formed upon its upper surface. A small steel pin *n,* is fixed to the end of the rod *m,* and is pressed against the heart by a drag-weight *o,* attached to it by a chain passing over a small pulley. As the roller revolves it produces a slow motion of the heart, causing the rod *m, m,* to traverse nearly the whole length of the boss, and thereby preventing the roving from wearing the surface of the rollers unequally.

q, q, the thread-plates or guides, having small notches opposite to each bobbin, through which the threads pass on their way from the drawing rollers to the eyes of the flyers. These plates are made in separate lengths as may be convenient, and are hinged in order to enable the bobbins to be inserted into or withdrawn from the spindles.

G, G, are sheet-iron linings extending from the beam C to

H, H, spouts formed under and within the rows of bobbins. These linings and spouts serve to collect and withdraw the water which is thrown off by the centrifugal force of the bobbins.

r, r, the flyers for guiding and winding the yarn upon

s, s, the bobbins, formed with a species of pulley on their lower flanges. In this machine the bobbins are not driven independently of the spindles, for a reason which we have specified in our introductory remarks. The natural tendency of the bobbins to wind on the yarn regularly is assisted by the following contrivance:—

t, t, are drag weights, attached by pieces of string to loops on the back of the bobbin-lifter; these cords pass across to a plate with a serrated edge, fixed to the front of the bobbin-lifter, and press against the grooves formed in the lower flanges of the bobbins. The friction thereby occasioned, which may be varied by changing the length of leverage at which the weights act, gives the bobbin the requisite retardation for winding on the yarn.

u, u, the bobbin beams or lifters supported by the traverse rods *x, x,* which are attached to bosses upon the traverse shaft *z, z,* by chains furnished with adjusting thumb-screws for adapting the bobbins to the height of the flyers.

v, v, are the spindle rails or beams, in which are inserted the steps and collars for the spindles to run in.

w, w, the spindles themselves, with their driving wharves or pulleys fixed to them.

I, is a bracket upon which works the strap-guide for starting and stopping the machine.

J, J, the fast and loose pulleys fitted to the end of

K, K, a long cylinder constructed of tin plates, extending the entire length of the machine, and forming a continuous drum for driving the spindles.

L, a balance pulley round which the tape passes for driving the spindles, and which keeps it at the proper tension. The tape passes over the cylinder K, then over the balance pulley L, and round two spindles on each side of the frame, thus causing one belt or tape to drive four spindles. Previously to the introduction of this method, each spindle was impelled by a separate tape.

M, N, O, P, Q, R, S, a train of spur geering, (O being a change pinion), constituting the twist geering, and conveying motion from the driving shaft to the front rollers on both sides of the frame.

T, U, V, a train of spur wheels situated at the middle of the frame, and constituting part of the traverse geering. T is a pinion fast upon the end of the shaft which carries the intermediate twist wheel P, and which has a bearing in the middle support B. This wheel works through the intermediate U, into V upon the mangle-pinion shaft.

W is the mangle-wheel, situated at the opposite end of the frame to the twist geering, and actuated alternately in both directions by the mangle-pinion.

X, a small spur pinion fixed to the axis of the mangle-wheel, and working into a rack formed on the top edge of

- Y, a cast-iron horizontal bar working transversely in slides bolted to the inside of the end framing A, A. Each extremity of this bar is formed into a radial rack; these work into the eccentric spur wheels *y*, *y*, fixed upon the traverse shafts *z*, *z*, imparting to the latter a *graduated* motion of rotation, which is communicated to the bobbin-lifter by the mechanism previously described; causing the flyer to wind the yarn upon the bobbin in a slightly spherical form.
- *a'*, *b'*, *c'*, *d'*, a combination of wheels forming the draught geering, which, being precisely similar to the draught geering in the other machines which have come under our notice, need not be further detailed. With the size of rollers and wheels given in our Plate, the draft of the spinning frame will be found to be 5·14. The twist is determined and calculated upon the same principles as that of the roving frame, substituting the diameters of the drum K, and driving wharves for the spiral wheels and pinions. Taking the sizes of these and the other constituent parts of the twist geering as represented in our Plate, the twist will be found to be 26·97 per inch.

PLATE CXXII.

STEAM ENGINE INDICATORS.

The important and useful little instrument which we have represented in this Plate has very materially contributed to the perfection and efficiency of our modern steam engines; not only by enabling the engineer to ascertain and register the exact values of the forces from which its power is derived, at the point where these forces come into effective operation, but also by pointing out the precise periods in relation to the different parts of the stroke, at which these elements of power come into action, and thereby conducing to the most economical and perfect combination of them. By its use he is introduced, as it were, into the interior of his engine, and is made cognizant of its most occult and delicate movements.

The idea embodied in this ingenious and beautiful instrument was originated by the justly celebrated James Watt, who, at a very early period in the history of the steam engine, employed a machine identical in the principle of its operation, though less compact in form than that now so extensively in use. His object was, to ascertain with certainty the mean steam pressure, and more particularly, the proportion which the vacuum in the cylinder bore, at different parts of the stroke, to that in the condenser, in order to determine the dimensions of cylinder required for any given power, as also the relative proportions proper to be given to the steam and exhaust ports, &c. Having attained these objects, and demonstrated by the unparalleled success which attended his engines, the judiciousness of the methods he employed to determine the proportions of their parts, succeeding mechanicians seem to have despised the unpretending little instrument by whose assistance he had been led to such results, and during the space of nearly half a century, to have trusted implicitly, in the construction of their engines, either to the absolute accuracy of Watt's data, or of their own theoretical deductions, in many cases extremely fallacious. From this state of oblivion, the indicator has been, at a comparatively recent period, rescued by the late Mr. Macnaught of Glasgow, who has greatly improved its construction, and put it into such a compact and portable form as to be easily applicable to steam engines of every description. Its consequent general adoption has led to some notable improvements, and materially elevated the standard of *duty* in steam engines; it has demonstrated the economy resulting from a liberal use of the expansive power of the steam, and the great advantage attendant upon a more rapid and complete exhaustion than could be attained by the arrangement of slide-valve previously employed.

MACNAUGHT'S INDICATORS.

The figures comprised under this head represent two different forms of the indicator, as at present constructed by Mr. William Macnaught: the first being that employed to measure the power of high pressure engines, and the second being more particularly adapted for those in which steam of a tension not greatly exceeding that of the atmosphere is employed. These two instruments differ in no respect from each other in the principle of their action, and but little in the details of their construction; accordingly, one description will suffice for both, in the course of which, we shall be careful to notice the points in which each respectively differs from the other. These drawings are very carefully executed from the machines themselves, and to a uniform scale of six inches to a foot, or half the real size.

General Description.

From a glance at the drawings it will be at once perceived that the indicator is neither more nor less than a small atmospheric steam engine, with the addition of a spiral spring and graduated scale to mark the degrees of pressure. The steam cylinder A is of gun metal, very accurately bored out and fitted with a solid piston *a*, ground into the cylinder so as to be perfectly steam-tight. A slender steel rod *b*, *b*, is screwed into the piston, and is guided at its upper extremity by passing through the cover *c*, of an external brass casing or cylinder B, screwed to the lower end of the steam cylinder. At this point is also fixed a stop-cock K, destined to form a communication between the cylinder of the steam engine and that of the indicator; in the high pressure indicator this cock is formed with a screw at its lower end, for the purpose of attaching it to the socket-cock, which is permanently fixed to the cylinder cover, while, in the instrument as adapted to low pressure engines, this attachment is considered to be sufficiently secure by the more convenient method of simply inserting the tapered end of the cock C into a corresponding socket accurately ground to fit it.

Towards the middle of the piston-rod *b*, *b*, a small stud *d*, is fixed immovably by soldering, and to it is screwed a brass disc, having the end of a spiral spring D soldered into it. The opposite extremity of this spring is secured, in the case of the high pressure indicator, to the steam cylinder A, and in that used for condensing engines, to the cover *c*, of the external casing. This difference in the arrangement of the springs is owing to the difference in the prevailing directions of the motion of the piston *a*, in either case; it being considered most expedient to make the springs act by distension rather than by compression. In the stud *d*, is a small square socket, into which, through a slot or opening in the external casing B, is inserted a steel piece, part of which is formed into an index or pointer, adapted to work against an adjustable graduated scale, screwed to the outside of the casing B.

The principle on which the strength of the springs is regulated and the scales are graduated is as follows: the area of the piston, the resistance of the spring, and the divisions upon the scale, are so proportioned to each other, that one pound of pressure in either direction upon the piston will cause the index to move through one division upon the scale. In the high pressure indicator, although the size of the piston is invariable, (being uniformly made equal to $\frac{1}{8}$ of a square inch in area), the length of the divisions upon the scale is arbitrary, being determined by the amount of steam pressure to which the machine may at any time be subjected, and by the length of scale that can conveniently be applied. The instrument represented in our Plate is adapted to indicate up to 60 lbs. of pressure, and the scale is equally divided into 20ths of an inch, each of these divisions representing one pound of pressure upon the square inch of the piston. From these data the spring is to be very carefully constructed, so that 2 ounces, (or $\frac{1}{8}$ of a pound), shall move the index through one division of the scale.

In the low pressure indicator the process is precisely the same in principle, though somewhat less involved. The tension of the steam being low, and the atmospheric pressure limited within 15 lbs. to the square inch, the scale is divided into 10ths of an inch. The piston is made equal to $\frac{1}{4}$ of a square inch in area, and the elasticity of the spring is such that 4 ounces, (or $\frac{1}{4}$ of a pound), acting upon the piston in either direction, will cause the index to move through one division of the scale, which, consequently, represents one pound of pressure upon the piston of the steam engine to which this instrument is applied. The zero-point is that at which the index stands when the cock C is shut and the piston *a* remains undisturbed, and therefore, when the instrument is in action, it denotes that point in the stroke at which the pressures above and below the piston are balanced.

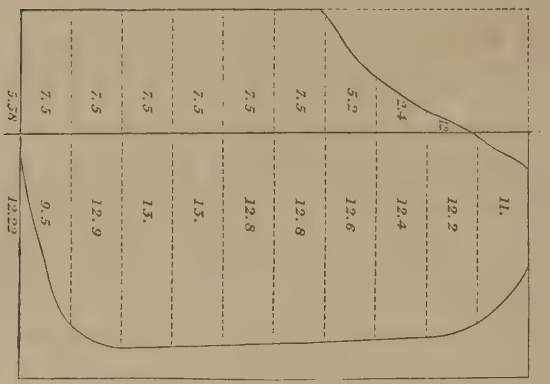
From these explanations it will be obvious that, by attaching the instrument to the cylinder of a steam engine, and observing the motion of the index upon the scale, the maximum steam pressure and vacuum may be at once ascertained. But this is not the only, nor even the most important function of the indicator. It was desirable to find out the exact periods and modes in which these two elements of power come into operation, and especially, the *mean* effective values of each; the rapidity of the motion through so short a space precluding the possibility of taking these observations with any degree of accuracy. These important objects are fully attained by the help of a simple and beautiful contrivance, by which the instrument is made to register its own performances.

An arm or bracket *g*, is firmly attached to the indicator by being clamped to the external casing B, on which it may be set to any convenient elevation, and there secured by a screw. To this bracket is riveted an upright axis *h*, on which, by a long socket, to ensure steadiness of motion, is accurately fitted a cylindrical piece F, formed into a pulley at its lower end. The other extremity of the socket carries a small cylindrical box containing a spiral spring *i*, (similar to the main spring of a watch), attached at one end to the fixed axis *h*, and at the other to the internal surface of the box in which it is inclosed. The bracket *g*, carries also a small friction pulley *j*, for the purpose of guiding to any convenient moving part of the engine a cord wrapped round, and attached to the pulley F; into which is screwed a small catch *k*, serving to circumscribe its motion to a single revolution. An external cylinder or drum E, which may be withdrawn from the instrument at pleasure, is fitted over the revolving cylindrical piece F, so as to partake of its motion, and upon it is fixed a slip of brass formed into a double spring *l*, *l*, for the purpose of securing the piece of paper on which the instrument is to register its performance. This is effected by means of a pencil *f*, placed in a holder *e*, jointed to the piece of steel on which the index or pointer is formed, and fitted with a small spring, so as to press the point of the pencil gently against the paper cylinder, or admit of its being withdrawn from contact with it at pleasure. From these arrangements it will be seen that if the piston *a*, be moved up and down while the pencil is in contact with the cylinder E, a straight line will be traced upon it in the direction of its length; and if, on the other hand, the cylinder be made to turn upon its axis by pulling the cord, while the piston remains at rest, a straight line will be traced round it at right angles to the former. By the combination of these two motions, when the instrument is in operation, a diagram is produced, which represents the performance of the engine at all parts of its stroke.

Action of the Instrument.—Fig. 9 represents the mode of applying the indicator to the cylinder of an engine. The cock C is inserted into the corresponding socket prepared for its reception, and the cord which passes under the pulley *j*, is attached to the radius bar, or other moving part of the engine, so as to cause the cylinder E to make one revolution on its axis, coincident with and representing the stroke of the engine; on the relaxation of the cord at the termination of the up stroke, it is taken up again by the action of the spring *i*, and the cylinder E resumes its original position. The slip of paper is then to be wrapped tightly round the cylinder, its ends being secured by the pressure of the two springs *l*, *l*. These arrangements having been made, the pencil *f*, is turned down into contact with the paper, and the engine is allowed to make a stroke or two with the cock C shut, so as to form an *atmospheric line*. A communication is then opened with the interior of the cylinder of the engine by turning the cock C, and a figure or diagram is traced upon the slip of paper, exactly representing the successive pressures of the steam above, and corresponding degrees of exhaustion below the atmosphere line at every part of the stroke. To find the mean effective values of each of these pressures respectively, the figure is to be divided, in the direction of its length, into any number of equal parts, the perpendicular distances of the outline of the diagram above and below the atmosphere line at each of these points is to be carefully measured upon the scale of the instrument, and the sum of these to be divided by the number of points taken. Hence the actual power of the engine is easily calculated.

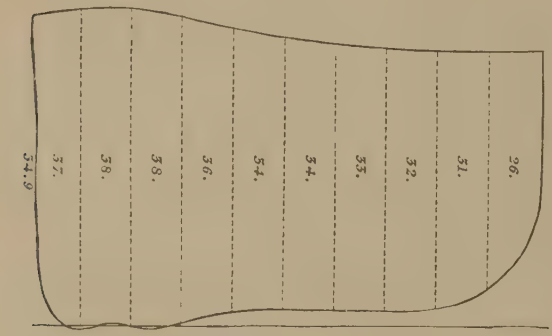
As an exemplification of these processes, we refer to the annexed wood-cuts. Fig. 1 is a fac-simile of a diagram

Fig. 1.



taken from a condensing marine engine, in excellent working order; working expansively, and showing a very large amount of gross effective pressure on the piston in proportion to the consumption of steam, (5·38 + 12·22 = 17·60). Further explanation of this figure is unnecessary, from the minuteness of our previous description. Fig. 2 is from a good high-pressure engine, receiving steam to the end of the stroke, and with a pressure of 40 lbs. in the boilers; the diagram showing a difference between the average pressure in the cylinder and that in the boiler, to the amount of nearly 5 lbs. In such engines, it is almost unnecessary to remark, the diagram is situated entirely above the atmosphere line.

Fig. 2.



MORIN'S INDICATOR.

That eminent French mechanician, M. Arthur Morin, conceiving with reason that considerable inaccuracy was likely to result from the difficulty of constructing the spiral springs in Macnaught's indicator, so as at all parts of the stroke to denote equal pressures by equal divisions; and, moreover, considering it desirable to have the means of taking indications throughout several consecutive strokes, has invented another description of instrument which we have represented in the remaining figures of Plate CXXII.

This indicator, like that we have already described, is adapted for being fitted to the cylinder cover of the engine; it carries a stop-cock pipe G, furnished with two keys; between these is situated a small horizontal cylinder H, in which a solid piston is accurately fitted to work steam-tight. Towards the middle of the piston rod *m*, which is properly guided to a rectilinear course, is a square part in which is inserted the lower end of a long parabolic spring *n*, the other extremity of which is fixed to the summit of a standard I, forming part of the frame-work of the machine; the spring being so adjusted as to allow the piston to travel a certain distance in both directions. The square boss of the piston rod carries also a small pencil *o*, for the purpose of tracing the different degrees of tension of the steam, on the opening of the lower cock G.

Two pencils *p*, *p*, are placed in holders fixed to the framing, exactly opposite to the point at which the pencil *o*, stands when the stop-cock G is shut, and being thus immovable, serve to mark a continuous atmosphere line. A third pencil *q*, which is destined to mark the termination of each stroke, is brought into contact with the paper by placing the instrument so that the working beam, cross-head or any other moving part of the engine may touch lightly, at the end of the stroke, the top of an upright rod *u*, which is connected by a system of levers *r*, *s*, *t*, with the top of the pencil *q*.

A continuous band or roll of paper may be subjected to the action of this machine for an indefinite period, so as to produce diagrams representing the action of the engine during several consecutive strokes. The roll of paper is first wound upon the cylinder L, by means of the handle *y*; it is then passed over the three small rollers *v*, *v*, *v*, placed to oppose the pressure of the pencils, and is received upon the cylinder M, situated at the opposite end of the framing Q, Q. On the axis of the cylinder M, is a conical pulley or fusee N, opposite to which is situated a cylindrical drum O, which receives a uniform motion from any rotating part of the engine by means of a worm-wheel *w*, on its axis, geering with an endless screw on the axis of the strap-pulley P. The cylindrical roller O communicates motion to the conical roller N by a cord wrapped round both, and fastened at opposite extremities of each. The object of this arrangement is to compensate for the increased surface velocity due to the increased diameter of the cylinder M as the paper is wound on to it, by imparting to it a proportionally retarded angular velocity.

This instrument, although highly ingenious in many of its details, and capable of giving very correct indications, is wanting in that portability and compactness which has very materially contributed to bring Macnaught's instruments into such general use. In the latter also, a considerable advantage results from the length of range, or distance through which the piston travels; the diagrams being made upon a large scale, the expert engineer is able at a glance, and without reference to the scale, to ascertain by the mere contour of the figure, whether the engine is performing all its functions properly.

SUPPLEMENT

TO THE

ENGINEER AND MACHINIST'S ASSISTANT.

PLATES CXXIII.—CXXVI.

JAMES NASMYTH'S PATENT STEAM HAMMER, MANUFACTURED BY NASMYTH, GASKELL & CO., PATRICROFT, NEAR MANCHESTER.

(COMMUNICATED BY THE INVENTOR).

Before proceeding to describe the principle, mode of action, and constructive details of the direct action steam hammer, it may be proper to make a few remarks on the ordinary forge hammers, so that the nature of the advantages possessed by the steam hammer may be more clearly understood.

In all forge hammers previously in use, the force necessary to set them into operation had to be transmitted in a very indirect manner,—for whether a water-wheel or steam engine were the moving power, the requisite lifting and falling action of the hammer had to be produced by the employment of rotatory motion, thus rendering necessary the use of wheels, shafts, cams and other cumberous details, which, together with the apparatus requisite to connect the various parts of the machinery, and give due strength and solidity to the whole, not only caused great outlay and sacrifice of valuable space, but also occasioned much loss of power, by reason of the very circuitous manner in which the force of the prime moving agent had to travel ere it reached its final destination, and came forth in blows from the forge hammer. Great inconvenience, also, was found to result from having a considerable portion of the working machinery close to the hammer, as thereby a very serious impediment was offered to the free execution of the work. And when we add to this the very limited range in the clear fall of an ordinary forge hammer, (seldom, in any case, exceeding 18 inches), causing the force of the blow to decrease in a very rapid ratio, with a moderate increase in the diameter or depth of the work; and when we take into consideration the fact that, in consequence of the helve of the hammer working on a centre or joint, its face is parallel to that of the anvil only at one particular distance; and finally, when to this list of inconveniences we add that in the ordinary forge hammer we possess no power or control over the force of its blows, but are compelled to make the best use we can of them, whether they be adapted to our purpose at the time or otherwise, we find inherent in the very principle of such hammers, a combination of evils and inconveniences that only excites surprise that they should have been suffered to exist for so great a length of time.

This remark is most strikingly applicable in the case of those forge hammers which receive their power from a steam engine, inasmuch as the power in question originates in the motion of the piston, in the very state and condition in which, for the purpose of hammering, we desire it ultimately to be, namely, as a straight up and down motion; so that instead of causing this reciprocating action of the piston rod to pass through all the complex media of beam, connecting rod, crank and cam shaft, for no other purpose than to cause it to act in the same manner as at first,—if we dispense with all this mass of intermediate machinery, and simply invert the steam cylinder so as to bring its piston rod out at the bottom of the cylinder, and attach it *directly* to a block of iron working in guides right over the anvil face, we shall then have obtained all the grand essentials of a forge hammer in its simplest and most obvious, and, at the same time, as experience has demonstrated, in its most perfect and efficient form. Such, then, is the Direct Action Steam Hammer, and from such considerations as the preceding was I led to the invention of this machine.

It may perhaps interest some of the readers of this work, to know the circumstances which led me to reflect on the subject of forge hammers with a view to their improvement, and which resulted almost immediately in the production of a rude sketch, in which are embodied all the main features, and most important details of an invention which appears destined to aid mankind in no small degree, in facilitating and improving the working of wrought-iron. These circumstances are as follow:—

On the 24th Nov. 1839, I received a letter from a mechanical correspondent, informing me that he had met with an unexpected difficulty in the carrying out of a grand project in which he was engaged, in the execution of which a paddle-shaft of upwards of 3 feet in diameter was required,—having found that there was not a forge hammer in England which could execute so large a piece of work with the requisite degree of solidity. The statement of this difficulty caused me to reflect on the nature of the construction of forge hammers, when by a rapid and happy train of reasoning or reflection, I saw before me in my mind's eye, the direct acting steam hammer, and in a few minutes I had it reduced to form in the shape of a rude sketch, which I have at present before me.

The great depression of business, more particularly in every thing connected with mechanical operations, that occurred during the years 1839, 40, 41 and 42, caused every endeavour to secure the co-operation of some practical iron master to aid me in bringing out this invention, to prove fruitless, as I found that those who were most likely to require such a hammer had not employment enough for those they already possessed; consequently, it was not until the revival of business in 1842 that I was successful in getting suitable parties to listen to my statements of the important advantages attending the use of the steam hammer, and in thus inducing them to come and witness its action at our works at Patricroft, near Manchester.

So clear, however, was the demonstration of the superiority of this steam hammer in point of simplicity, efficiency, and facility of management, that it at length attracted the attention of parties who had the power to give it a trial on a considerable scale. The first who did so were the Lowmoor Iron Company, who gave an order for a hammer of 2¾ tons, with a clear fall of 6 feet; this was started, after considerable delay, in December, 1843. The next party who ordered a similar hammer was Messrs. Rushton and Eccersley of Bolton, to whose enterprising and hearty zeal in bringing out this invention under the most favourable circumstances, I feel especially indebted. The high success which attended their use of this steam hammer very rapidly cleared away all that prejudice which so generally attends the introduction of any new machine. A similar hammer, ordered about the same time by Messrs. Hawkes, Crawshaw and Sons of Newcastle-upon-Tyne, was equally successful, having met with the same unprejudiced treatment as at Bolton, and yielded the most satisfactory results in consequence. These prominent examples of success had their natural effect in aiding the progress of the invention; and its merits having been speedily recognised by the Admiralty, an order was received to erect one of 2½ tons at the Royal Dock Yard at Devonport. Most distinguished success attended on this hammer, which was seen on several occasions by the Lords of the Admiralty, and by parties from several foreign governments; the result of which was an order to erect a set at each Royal Dock Yard, and similar orders for several of the Arsenals of continental governments. But what is most gratifying is the rapid progress it has made, and the high estimation in which it is held by the proprietors of almost all the principal private establishments throughout Europe and the United States; many of them having extended their orders for second and third hammers, in consequence of the satisfaction they have experienced in its performance; having found it most useful in facilitating and perfecting the execution of every description of large wrought-iron work, as well as in the several early stages in the manufacture of wrought-iron, as in hammering *puddled balls*, shingling, and welding slabs for the best description of boiler plates, &c.

Some idea of the efficiency of the steam hammer in shingling puddled balls may be formed from the fact, that one of 30 cwt., which has been for nearly two years in constant operation at the Gartness Works of the Monkland Iron Company, in the West of Scotland, works off with perfect ease the constant produce of from 18 to 20 puddling furnaces. For this duty the steam hammer is found to be peculiarly adapted, as it can be made to act for the first few strokes as a *squeezer*, to bring the puddled ball to a neat cubical form; after which it may be made to deal out upon it such energetic blows as secures the entire expulsion of all cinder and other non-metallic impurities, the absence of which, to a greater or less extent, mainly determines the quality of wrought-iron. In short, in every process where either blows of the most enormous energy, or slight taps of the utmost gentleness are required, either continuously or in all grades of variation from the one extreme to the other, the steam hammer offers facilities which have never hitherto been obtained from any mechanical contrivance for such purposes.

Enumeration of the Plates.

Plates CXXIII. CXXIV. Fig. 1 represents a side elevation of the steam hammer, exhibited in full operation, the hammer-block, valve geer and other working parts being disposed in the positions which they occupy at the termination of a stroke.

Fig. 2 is a general plan corresponding to the above.

Plates CXXV. CXXVI. Fig. 3. An end elevation, and fig. 4, a vertical transverse section of the machine.

Fig. 5 is a sectional elevation of a portion of the machine, showing the positions of the hammer-block, valve geer, and other working parts when the hammer is raised for a fresh stroke.

Fig. 6 is a sectional plan of the steam cylinder and its adjuncts, taken on the dotted line 1—2 in fig. 4.

Fig. 7 is a sectional plan of the steam passages, &c. on the dotted line 3—4 in the same figure.

Fig. 8 a sectional plan on the dotted line 5—6, and fig. 9, a sectional plan on the dotted line 7—8.

General Description.

The framing of the steam hammer consists of two strong cast-iron standards A, A, bolted and further secured by keys to a broad base-plate B, B, embedded in the solid masonry forming part of the floor of the forge.* The standards are surmounted, and their upper extremities united by a species of entablature C, in which the steam passages and valve face are formed, and to the upper surface of which the steam cylinder D is bolted. The piston-rod E is fitted to work vertically through a stuffing-box in the centre of this entablature, and its lower extremity is directly attached to the mass of cast-iron F, forming the hammer-block, which is guided to a strictly vertical and rectilinear course by being made to work freely in planed guides formed on the interior surfaces of the standards A, A. The hammer

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* In those localities where either masonry would be very expensive, or where the soil is of a loose nature, (as in the case of *forced* or *made ground*), the most economical and efficient foundation for the anvil-block and base-plate is formed by driving piles, and resting the anvil-block on the heads of them. Several foundations for the Steam Hammer have been made in this way, and have given the greatest satisfaction, being remarkably free from vibration.

a, itself is inserted into a dove-tail recess in the bottom of the block F, where it is retained by wooden packing and iron wedges; while the anvil *b*, is in a similar manner secured to the anvil-block G, which is a mass of cast-iron of such weight as effectually to oppose, by its inertia, the momentum of the hammer, and prevent the force of the blows from being dissipated.

Such are the main features of this machine; from which it will be at once understood that, if we can provide the means of rapidly raising the hammer-block to a sufficient elevation, and then as rapidly letting it fall down upon, and so give a blow to the work placed upon the anvil, we have all that is requisite to produce a forge hammer in its simplest, and, at the same time, its most powerful and perfect form. I shall proceed to describe the means by which this is accomplished, and the mechanism by which the rapidity and intensity of the blows can be regulated at pleasure, and the machine rendered automatic or self-acting.

The duty above adverted to, of raising the hammer-block, is performed by the direct application of the elastic force of steam. For this purpose, the steam is led on to the machine by the steam-pipe H, communicating with a neighbouring high-pressure boiler; a throttle, or shut-off valve *c*, enclosed within the valve box I, being situated close to its junction with the main steam-valve chest J, and brought within the control of the attendant workman by means of the rod and lever *d*, *d*. The alternate admission of the steam into the cylinder by the port *f*, and its escape therefrom by the passage *g*, and waste steam-pipe K, are regulated by means of the slide valve *e*, which may either be worked by hand, or, through the intervention of the self-acting mechanism to be hereafter specified, by the action of the machine itself. The piston L, which is strongly constructed of malleable iron, and fitted with a single packing ring, works steam-tight within the cylinder D; and being directly attached by the piston rod E to the hammer-block F, it will be obvious that, on the admission of steam of sufficient elastic force beneath the piston, we are supplied with the means of raising the hammer-block to any required height within the range of the machine; while by opening the communication between the under side of the piston and the external atmosphere, the action of gravity will be unimpeded, and the hammer will descend upon the work placed on the anvil, and discharge a blow upon it, energetic in proportion to the weight of the hammer-block, and the height from which it has fallen.

And as, by these simple means, there is no practical inconvenience in supplying the power to raise a hammer-block of 5 or 6 tons weight to an elevation of 7 or 8 feet above the anvil, some idea may be formed of the vast energy of the blows given out by such a mass of iron falling rapidly through so great a space, and discharging *the whole** of its momentum upon the work placed below on the anvil to receive it. It will be seen, further, that the anvil face and hammer face are at all times parallel to each other, whatever be the height or distance between them. The practical value and importance of this property, which is inherent in the principle of the steam hammer, has been duly appreciated by all who have had experience of the working of this machine.

With a view to prevent any risk of the piston striking the cylinder cover when working to the full, or very highest stroke of the hammer, a very simple but effective air or steam recoil spring is provided, by having the cylinder cover screwed down quite air-tight, so that as soon as the piston passes, in its upward motion, the holes *h*, *h*, the air or steam remaining above is shut up in the upper chamber; and as it has no means of escape, it acts as a most perfect spring in arresting any further rise of the piston; and has, besides, the important advantage of converting into increased *downward* velocity of action the undue upward action, which might otherwise have proved not only useless but destructive. The increase of energy in the blows which can be obtained by this simple means is a point of considerable importance. It is scarcely necessary to remark that, in the emergency above adverted to, the apertures *h*, *h*, act as safety valves for the issue of the main body of the steam, which escapes through the passage *i*, *i*, into the exhaust or waste steam-pipe K.

Another point of constructive detail worthy of special notice is, the peculiar mode which I have adopted for connecting the piston rod to the hammer-block. This is one of the most important details in the entire invention, and without which no practical success would have attended it. Had the piston rod been attached to the hammer in the ordinary mode of attaching pistons to the machinery of a steam engine or such like, namely, by a cotter, or by screwing the rod into the hammer, or such other solid, unyielding mode, the effect of the blow or fall, at each stroke of the hammer, would have been that the piston rod and piston, (being composed of a considerable *mass* of materials), would have themselves acted *as a hammer*, and would have discharged their momentum upon the means of fastening, and this with such destructive effect as to break through all such *solid, unyielding means of resistance*, after a few blows.

This was foreseen from the first as an action to be prevented, and accordingly, in my original drawing, already adverted to, a remedy was provided, which experience has proved to be entirely effectual.

This contrivance consists in placing, in a cylindrical recess formed in the body of the hammer-block, and under the knob *i*, on the end of the piston rod, a series of pieces of hard wood, or other *slightly* elastic material, (see fig. 4.) The effect of this arrangement is to allow the momentum of the piston and piston rod to deposit itself in such a comparatively gradual manner as to cause the concussion arising from the most severe and energetic blows of the hammer to have not the slightest evil effects on the piston and rod; it is, in fact, the very same expedient to which nature has had recourse for the purpose of obviating those unpleasant and destructive shocks and vibrations which we should experience at every step or stamp of the foot, had no cartilage been provided between the joints of our bones. It is surprising to observe by how small an amount of elasticity, from the employment of such compressible material, the evil effect of violent shocks may be removed. The connection of the piston rod and hammer-block is secured by means of the two keys *k*, *k*, driven very firmly *above* the knob or button *j*, a layer or two of the elastic material being interposed for the purpose of neutralising any shock in the contrary direction.

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* In the case of the old system of Forge or Helve Hammers, about one-third only of the total weight of the hammer was effective, the other two-thirds resting on the pivot-standards; so that, in this point of view, the proportion between the blow of a Steam Hammer and that of a Helve Hammer is nearly 3 to 1 in favour of the former, to say nothing of other advantages.

I shall now proceed to describe the mechanism by which the height of the fall of the hammer, and consequent intensity of the blow, may be modified according to circumstances, and the machine made perfectly self-acting.

The requisite alternating motion of the steam valve *e*, is produced in the following manner:—The valve spindle *l*, is prolonged upwards and attached to a small solid piston *m*, working within a short cylinder M, bolted to the main steam cylinder D. A small portion of steam is supplied above the piston *m*, by a slender copper tube *n*, communicating with the steam-valve chest J; by this arrangement it will be seen that, unless counteracted by some superior force, the pressure of the steam upon the piston *m*, will tend to keep the valve *e*, constantly depressed, in which position the steam-port *f*, is full open. This counteracting force is supplied by the action of the hammer itself; for, by means of the tappet N, (which is bolted to the hammer-block), coming into sliding contact, when the latter is raised, with the small friction roller *o*, mounted on the end of a bent lever O, O, the screwed rod P, which is jointed to the opposite end of that lever, is depressed, and that motion being communicated to the valve-spindle *l*, through the intervention of the connecting rod Q and valve lever R, the steam valve *e*, is raised, thus cutting off all further ingress of steam under the piston, and almost at the same instant permitting the escape of that which had served to raise the hammer. By this simple contrivance the upward motion of the hammer is made the agent for its own control in that respect. By comparing the relative positions of the parts referred to, as exhibited in figs. 1 and 5, the nature of the motion above described will be at once most fully understood. To obviate the injurious effects of the shock of the tappet N against the lever O, a connection is provided at *p*, on a similar principle to that formerly described in reference to the connection of the piston rod and the hammer-block; and in order to restrict the downward travel of the valve to the proper point, a check or *buffer-box* S is provided, consisting of a small cylinder bolted firmly to the framing of the machine, within which a circular nut, screwed on the lower end of the rod P, works as a piston, a few leather washers being interposed between the latter and the close or upper end of the cylinder.

It may be here remarked, that it is by no means necessary to continue the admission of steam under the piston until the termination of the upward stroke, or *lift* of the hammer, seeing that the velocity which the hammer-block has acquired in its upward motion makes it continue to ascend after the further ingress of the steam has been arrested. This circumstance is a source of considerable economy of steam, as we have by such action, (as well as by that due to the expansive energy of the steam), an effect as to height of lift of the hammer, greater than that which is due to the actual expenditure of steam at its original pressure. It is worthy of remark, also, that as the *over-running* action above alluded to will necessarily increase in proportion to the velocity of ascent of the hammer-block, this circumstance will, to a considerable extent, compensate for the increased expenditure of steam due to that increased velocity.

From the above description it will be obvious, that the lift of the hammer, and consequent intensity of the blows, depends simply upon the position of the lever O, in relation to that of the hammer-block when at its lowest point. Therefore, if we can provide the means of altering the distance between these two points, we shall have it in our power to modify permanently the force of the blows to any required extent within the range of the machine. This condition is most completely satisfied by the arrangement of mechanism employed by me, and which is clearly represented in figs. 1, 5, and 8.

The rod P, which conveys the action of the lever O to the valve lever R, is screwed throughout the greater part of its length, and is so adjusted in its bearings, as to be susceptible of rotatory as well as vertical motion. This motion of rotation is imparted to it by means of a handle fixed to a short axis, working in a bracket T bolted to the framing, and actuating a pair of small bevil wheels *q*, *q*. The nut through which the screw works forms the point of attachment between the rod P and the lever O, the connection being effected by means of a short intermediate rod for the sake of ensuring parallelism of motion. A pair of small spur-wheels *r*, *r*, (through the first of which the rod P works by means of a sunk feather), serve to transmit the angular motion of the rod P to a similar screwed rod U situated parallel to and at a short distance from the former; the nut of the screw U forms the fulcrum or centre of motion of the lever O, and the pitch of the threads of both screws being equal, though formed in contrary directions to each other, it is obvious that, on turning the handle, the lever O and all it appendages will be simultaneously raised or depressed, and consequently the lift of the hammer regulated to any required extent, and its amount altered with the utmost ease and precision. The pin which forms the centre of motion of the lever O is protected and secured from lateral strains, by the cast-iron guides V and W, seen most distinctly in the sectional plan fig. 8.

A most essential part of the self-acting geer remains yet to be noticed. It is obvious that, were no provision made for the retention of the steam-valve in the position into which it is thrown by the upward motion of the hammer-block, the latter would not be permitted to have its due effect in the accomplishment of its work; for, as soon as it descended so far as to relieve the end of the lever O from contact with the tappet N, the valve would resume the position into which it is constantly solicited by the action of the *steam spring* at M, and the descent of the blow would be impeded by the return of the steam into the cylinder, before the hammer had completed its fall. To obviate this inconvenience, a simple but most effectual contrivance has been applied. Towards the lower extremity of the valve-screw P, a shoulder is formed, against which a short lever *w*, called the *trigger*, is constantly pressed by the spring *x*, so that when the rod P is depressed by the action of the lever O, it is arrested by the trigger and retained in that position until the blow has been struck. This delicate and most important part of the mechanism is very carefully constructed, the point of the trigger, and the shoulder against which it acts, being formed of steel, and hardened to resist wear.

To release the valve-screw from the trigger, and so permit the return of the valve into the position requisite for effecting a fresh stroke, the following mechanism has been adopted; on the front of the hammer-block, (see figs. 1 and 5), a lever X, called the *latch-lever*, is fitted to work freely on a pin passing through the body of the hammer-block. That portion of the latch-lever which is most remote from the valve geer is considerably heavier than the opposite end, and is constantly pressed upwards by means of a spring. The lighter end is brought into contact with a long bar *s*, *s*, called the *parallel bar*, the extremities of which are suspended upon two small bell-cranks *t*, *t*, whose other arms are connected by means of a slender rod *u*, (see fig. 3), forming a species of parallel motion, for the purpose of adapting this geer to come into efficient operation, at whatever point in the range of the hammer its blow may be arrested. A small connecting rod *v*, between the lower bell-crank and a short lever on the axis of the trigger *w*, completes this part of the mechanism.

The action of this geer is of a very peculiar nature, and is admirably adapted to fulfil the object intended. At the instant the hammer gives a blow to the work upon the anvil, the effect of the concussion is to cause the momentum of the heavy end of the lever X to overcome the upward pressure of the spring, and thereby to protrude its opposite end against the edge of the parallel bar *s*, which motion, though but slight in amount, is yet adequate, through the arrangements above described, to throw back the trigger from contact with the valve-screw, and leave the latter free to obey the impulse of the steam spring in the re-adjustment of the valve into its original position.

These various movements, which have taken so long to describe, are all performed in less than half a second, and consequently the action of the hammer is proportionally rapid.

The construction of the self-acting geer is so arranged as to admit of advantage being taken, when circumstances render it desirable, of the very action to obviate which the trigger *w*, is introduced. When it is desired to strike a gentle blow, such as is frequently required during particular stages in the progress of a piece of work, it is not requisite, for this purpose, to change the position of the valve lever O. All that has to be done is to hold back the point of the trigger *w*, by its handle *y*; this permits the valve to re-open and let the steam in under the piston L, at the instant the tappet N has fallen away from contact with the lever O. The effect of this is, that a quantity of steam is admitted into the cylinder under the piston, which serves as a cushion, by which the violent fall of the hammer is arrested, and its momentum modified to any extent, or at the pleasure of the person in charge of the handles. The handle *z*, is for the purpose of placing the steam-valve also under his control, and, for his further convenience in the management of the hammer, a platform Y and hand-rail Z are erected against the framing of the machine.

JAMES NASMYTH, Bridgewater Foundry,
Patricroft, near Manchester, February, 1847.

PLATES CXXVII., CXXVIII.

WROUGHT-IRON FURNACE FOR JAMES NASMYTH'S PATENT STEAM HAMMER.

These Plates represent minutely and accurately the form and dimensions of a most important adjunct to the steam hammer; with slight modifications it will be found applicable in many stages of the Iron Manufacture. The example chosen is one of a set which Mr. Nasmyth is now constructing for the Royal Dockyards, and other establishments, where he is at present engaged in erecting his steam hammers. Their principal peculiarity consists in their being bound together by plates of sheet-iron instead of cast-iron, whereby the danger of cracking and warping from the intensity of the heat is obviated.

Enumeration of the Parts.

A, A, the hearth and building upon which the furnace is erected.

B, B, the roof and sides of the furnace, also formed of fire bricks. The roof is arched throughout its entire length, in order that the heat may be reflected and concentrated on the work placed on the hearth.

C, C, the sheet-iron sides of the furnace, by which the brick work is secured and retained in its proper form.

D, D, the end plates for binding the side plates together. Instead of being riveted to the side plates, they are secured by bolts and nuts, so that the whole structure may be easily taken asunder when it is necessary to rebuild the furnace.

E, the ash-pit.

F, F, cross-bearers of wrought-iron for supporting the furnace bars *a*, *a*, *a*. The ends of the bearers rest in small cast-iron brackets *b*, *b*, secured to the sides of the ash-pit.

G, the passage to the chimney, formed in continuation of the arch of the roof.

H, H, the chimney, constructed internally and throughout its entire height, of fire brick.

I, I, corner pieces of ordinary bricks, built upon the angles of the interior chimney for the purpose of giving stability to the whole structure, which is further bound together by bolts *d*, *d*, *d*, passing through the small cast-iron pieces *c*, *c*, *c*.

J, J, a cast-iron sole-plate for supporting the brick-work of the chimney, and which is itself supported by

K, K, four strong cast-iron columns resting on a solid foundation of mason-work.

L, the stoke-hole through which the fuel is introduced into the furnace. The mouth of this stoke-hole is so constructed as to admit of its being stopped with a piece of coal when the furnace is in full operation.

M, a small square aperture in the side of the furnace by which the attendant is enabled to inspect the state of the furnace without interrupting the progress of the work. It may be stopped with a single brick.

N, N, the main openings into the furnace, through which the shaft or other work to be heated is passed.

O, the sliding door by which the aperture N is guarded. It consists of a square cast-iron frame, lined internally with fire bricks, and fitted to slide vertically between guides of angle iron.

P, P, levers working upon the cast-iron brackets Q, Q, surmounting the furnace. They are loaded at the outer ends with counterweights, and attached by short connecting rods to the doors O, O, so as to enable the stoker to raise or lower the latter with the utmost facility.

R, a register or damper surmounting the chimney, for the purpose of regulating the draught of the furnace. It is brought within the command of the attendant workman by means of a long chain or wire *e*, *e*, *e*, depending from the lever upon which it is hung.

PLATES CXXIX.—CXXXII.

JAMES NASMYTH'S PATENT STEAM PILE DRIVER, MANUFACTURED BY NASMYTH, GASKELL AND CO., PATRICROFT, NEAR MANCHESTER.

(COMMUNICATED BY THE INVENTOR).

At the present period, when so many important works in Civil Engineering are either in progress or about to be undertaken, the Steam Pile Driver has excited a considerable share of attention in the mechanical world, from the unprecedented facilities which it offers for the execution of such works, and from the reports of its extraordinary efficiency and superiority over the old system, in those instances where its performances have been put to the test. I feel confident, therefore, that the following detailed account of its construction, and remarks upon the principle of its action, will prove highly acceptable and interesting to most readers of this work.

The stability of many important public works mainly depends upon the manner in which the piles forming their foundation are driven, and instances of their failure from imperfect piling are not unfrequent, and when they do occur, entail ruinous expences upon the parties interested. In the year 1842 it occurred to me that the principle of my direct action steam hammer might be applied to supersede the old tedious and imperfect mode of driving piles, and accordingly, I patented this application of the principle, and on the receipt of the first order, I produced a complete Pile Driving Engine, whose performances far exceeded my most sanguine expectations. Such is the ease, accuracy and rapidity with which this machine goes through its work, that it may be confidently predicted, not only that it will soon be universally employed in all operations of ordinary magnitude where piling is required, but that it will ultimately lead to the undertaking of important public works, which from the great cost and difficulty of execution by the old method, must have remained practical or commercial impossibilities. By the use of the Steam Pile Driver, such works as breakwaters, and embankments of the sea and at the mouths of rivers, for the recovery of land and other purposes, may now be executed on the grandest scale, and at a comparatively moderate cost.

Although this machine has been in practical operation only since the autumn of 1845, the brief history of its progress during that period has been of such a satisfactory nature as to demonstrate most fully its capabilities, and to argue favourably for the extent and utility of its future applications. It has already been successfully employed in some of the most gigantic works in Europe, America and Africa. In the latter country two of these machines are at present engaged in the work of damming up the Nile at Cairo, for the purpose of rendering Lower Egypt independent of the annual inundation of that river, by furnishing the means of irrigating the corn lands of that country *at all seasons;* thus it is anticipated with reason, that Egypt will be rendered permanently, as it has hitherto been periodically, one of the most fertile countries in the earth. A more striking instance could not be adduced of the vast benefits which this machine is calculated to confer upon the world at large.

There are few departments of mechanical science of more practical importance than that which furnishes us with correct ideas as to the conditions and modifications under which mechanical force must be applied, in order to perform certain duties. The truth of this remark is strikingly exemplified in the case of driving piles, as I shall endeavour to explain.

Although theory demonstrates that we have equivalent mechanical forces in the case of

4 lb. falling with a velocity of 4000 feet per second, or
4000 lb. falling with a velocity of 4 feet per second,

these two being said to be theoretically equal, as regards the resulting momentum, yet how very dissimilar are the effects of each, when employed to perform such duties as that of driving a pile!

In the case of the first, namely, the 4lb. at 4000 feet per second; were we to attempt to drive a pile by depositing on its head momentum under such conditions, we should find that while we failed to drive the pile one inch into the ground, we should succeed in shattering it into splinters; simply because we employ the momentum in an unfit condition for the performance of the required duty; in such a case as this we have *destructive* instead of *impulsive* energy. The reason of this is that sufficient time is not allowed for the transmission of the force throughout the mass, so that the particles we thus attempt to move forward are crushed into each other, and splitting, rending and shattering is the result, instead of the required motion of the entire mass through the resisting medium into which we desire to impel it.

Now let us consider what is the result by the employment of the second named *equivalent force*, namely, the 4000 lb. travelling with a velocity of 4 feet per second. Here we have only 1-1000th part of the *velocity*, while we have 1000 times the mass. The effect of a blow on the head of a pile from the resulting momentum in this case would be, that while the head of the pile was not in the slightest degree injured, it would receive motion throughout its entire mass, and be driven several feet into the soil at each blow. Exactly such is the action of the Steam Pile Driver, as compared with that of the ordinary machines. Instead of attempting to obtain high momentum from a small mass of iron falling from a great height, I employ the momentum resulting from a great mass falling with moderate velocity from a small height, (3 feet), and instead of having one blow per minute, I employ from 70 to 80 blows per minute. The result is, that while the pile is driven with ease and rapidity into the most rigid and resisting soil, the head of the pile is so little injured as to present a neater appearance after having been driven than at the commencement of the action. All this advantage arises simply from employing the mechanical force in its proper condition for the performance of the required duty. If we desire to split, rend and shatter to splinters, let us give the blow from a small mass travelling at the highest possible velocity; if on the other hand, we desire to propel and push forward such a mass as a log of timber, let us do just the reverse, namely, give it a blow from a heavy mass moving at a low velocity.

Enumeration of the Plates.

Plates CXXIX., CXXX., Fig. 1 is a front elevation of the entire machine, shown in full operation driving a pile.

Fig. 2 is a corresponding general side elevation.

Plates CXXXI., CXXXII. Fig. 3 is a general sectional plan of the machine.

Fig. 4 is a transverse section of the stage or platform, taken on the dotted line 1—2 in fig. 3.

Fig. 5. An enlarged elevation of the steam-chest and safety-valve geer.

Fig. 6. A section corresponding to the above.

Fig. 7 is an enlarged section of one of the joints of the flexible steam-pipes, for conveying steam to the hammer cylinder.

Fig. 8. A plan corresponding to the above.

Fig. 9. A sectional elevation of the hammer cylinder and pile-case with all their appendages. In this view the hammer is supposed to have just effected a blow upon the head of the pile, and the various parts of the valve geer are represented in the positions they occupy at the commencement of a fresh stroke.

Fig. 10. A front view of the hammer cylinder and pile-case with their various attachments.

Fig. 11. A sectional elevation of the same parts as are represented in fig. 9, with the driving apparatus and valve geer shown in the positions they occupy when the hammer is about to fall.

Figs. 12 and 13. Enlarged views of the trigger and parallel motion of the valve geer detached. These views are drawn to a scale of twice the size of the other figures in this Plate.

Fig. 14 is a section of the hammer-block and hammer.

Fig. 15 is a sectional plan of the pile-case, taken on the dotted line 3—4 in fig. 10.

Fig. 16. A sectional plan of the pile-case and hammer-block, taken on the dotted lines 5—6 in figs. 9 and 11.

General Description.

There are two important features of novelty which serve to distinguish my Patent Steam Pile Driving Engine from all such as had previously been employed for the same purpose. These consist: First, in the direct manner in which the steam is employed as the agent by which the block of iron which strikes the head of the pile is raised to the required height; and, Secondly, in the peculiar mode in which the pile, while being driven into the ground, is employed to support that part of the apparatus which is directly concerned in driving or forcing it into the soil—the apparatus in question being so contrived that, as the pile penetrates into the ground, the superincumbent machinery shall follow down with it, until it has reached the required depth.

On the first of these peculiarities I need not here dwell, having already fully elucidated the principle in reference to my direct action steam hammer; suffice it to say, on this point, that as the nature of the work to be performed is in both cases precisely similar, the same arguments which were employed to prove the advantages attending the adoption of this principle in the case of the steam hammer are fully applicable in the present instance. By the peculiar arrangement referred to under the second head, we secure two most important practical advantages: First, the piles are guided throughout their entire course with the utmost accuracy and precision, so that no one shall twist or swerve in the slightest degree from the general line, or from the position in which it is set at the commencement of the driving process; (this object being attained without any sacrifice of power from the friction of bands, or other appliances usually employed for that purpose); and Secondly, the entire dead weight of the driving apparatus is brought into effective action to second and follow up the blow last inflicted upon the head of the pile. This last peculiarity of action constitutes one of the most important features of my machine, and has contributed very materially to the success which has attended its use. During the brief intervals between the blows, (70 to 80 per minute), the pile is urged continuously downward by the action of the force here adverted to, so that it may be said to be in a state of constant motion from the commencement to the termination of the driving, and thus the power which would otherwise be wasted in overcoming its inertia is profitably expended in forcing it farther into the soil.

The basis or foundation on which the entire machine is erected consists of a strong wooden stage or platform A, A, firmly framed together and strengthened by diagonal timbers, and wrought-iron corner pieces, the whole being further secured by the massive cast-iron brackets B, B, at each angle of the platform, in which the locomotive wheels *a*, *a*, *a*, *a*, are fitted to work upon rails disposed parallel and close to the line of piling on which the machine is destined to operate. The stage is made of sufficient dimensions and weight to give the requisite stability and firmness to the entire structure, and to afford room for the steam boiler, the workmen, and all that is necessary for the accomplishment of the proposed object. The great vertical guide-pole C, C, on which the driving apparatus slides, is securely bolted to one side of the platform, the boiler being situated towards the opposite side, to counterbalance the weight of the former, and to afford an abutment for the diagonal timber supports D, D, firmly bound to both by plates of iron and numerous bolts; the entire frame-work of the machine is further secured by the four adjustable tie-rods or stays *b*, *b*, *b*, *b*, attached to the four corners of the stage and to the top of the upright. This latter is surmounted by a cast-iron socket-frame, supporting the brackets E, E, which carry a chain-pulley, over which works the great chain *c*, *c*, one end of which is passed round a barrel worked by a small steam-engine, as will be hereafter described, while the other end is attached to, and sustains the weight of, the pile-driving apparatus.

This consists of a steam cylinder F, with all the necessary appendages of piston, valves, &c., as will be more particularly specified below; the lower flange of the cylinder is firmly bolted to the *pile-case* G, G, which is a species of rectangular box of a square section, constructed of plates of wrought-iron, strongly framed together in the manner indicated in the detail views in Plate CXXXII. The interior surfaces of the pile-case serve to guide the hammer-block in its vertical motion, and it is itself guided along the great upright C, C, by the pieces *d*, *d*, which are fitted to embrace the projecting slips of iron *e*, *e*, bolted to the front of the upright throughout its entire length. The lower end of the pile-case is open, to admit the head of the pile, and is furnished with cast-iron jaws or resting pieces *f*, *f*, (see figs. 9 and 11), bolted to its interior surfaces; these are so formed as to rest upon the shoulders of the pile H, which, if we suppose the great chain-barrel to be left free to revolve, thus becomes the sole support for the weight of the whole mass of the driving apparatus. By these arrangements it will be seen that as the pile is, by successive steps, forced into the ground by the action of the hammer, (the chain barrel being thrown out of geer with its driving apparatus during the process), the pile-case with all its appendages, weighing about 3 tons, is left at perfect liberty to bear upon the shoulders of the pile, and follow down along with it, while at the same time, and by the same means, the pile itself is guided into a strictly vertical and true course.

The driving apparatus consists simply of a modification of my steam hammer, the action of the various parts being in all respects identical, though the whole is movable, as above described, instead of being fixed. The steam necessary for its supply is generated in a boiler I, the construction of which is very similar to that of the ordinary locomotive engine boiler, as minutely detailed in Plates LXVI—LXXIII. of this Work. The steam-chest J surmounts the fire-box, and is made of sufficient height to prevent the influx of water into the steam-pipes, and the entire boiler and steam-chest are covered externally with a coating of felt, and with strips of wood to prevent the radiation of heat, and to give greater symmetry of appearance. On the cover of the steam-chest, (the internal construction of which is fully shown at fig. 6), is cast a small square box *g*, containing suitable bearings for the safety valve, which is loaded by a weight and combination of levers *h*, *h*, *h*, and for the throttle or shut-off valve, commanded by the lever-handle and rod *i*, *i*. The steam is conveyed from the boiler to the valve-chest of the driving cylinder, by a flexible steam-pipe K, K, composed of several lengths of wrought-iron tube, connected together by swivel joints of cast-iron *j*, *j*, *j*, the construction of which will be fully understood by reference to figs. 7 and 8. This arrangement admits of the steam-pipe accommodating itself, without any loss of steam from leakage, to every variety of height or distance at which the driving cylinder may be from the boiler, from the commencement of the process, when the apparatus is sitting aloft upon the shoulders of a tall pile, until it has arrived at its lowest position, when the pile has penetrated the soil to the required depth.

The remaining part of the driving apparatus is, as I have before had occasion to remark, identical in the principles of its action, and very similar in the details of its construction, to those of my direct action steam hammer. Referring, therefore, for fuller information on these points, to my description of that machine, I shall content myself, in this place, with giving a mere enumeration of the parts.

F is the steam cylinder, within which the power necessary for raising the hammer to the required elevation, (3 feet), is generated.

L, the steam-valve chest, bolted to the lower side of the cylinder, within which the valve *k*, is fitted to work upon a face cast with the cylinder. The steam, after having accomplished its work, is permitted to escape into the atmosphere by an oblong aperture *l*, formed in the cylinder face; and, to obviate the risk of accident from the piston rising too high, a number of small round holes *m*, *m*, are formed near the top of the cylinder, so that the steam may blow out into the air when the piston rises above their edges. It may here be remarked that the efficacy of the blows of the hammer, and the security from damage to the parts of the machinery, are in this case, as in that of the steam forge hammer, materially augmented by the recoil of the air or steam enclosed above the piston.

M, the piston, formed of wrought-iron, and fitted with a single packing ring.

N, the piston rod, having a cylindrical boss or enlargement *n*, at its lower extremity, for the purpose of affording means for securing a slightly elastic connection, by hard-wood washers, between the piston-rod and hammer-block.

O, the hammer-block consisting of a rectangular mass of cast-iron weighing 30 cwt., adapted to slide freely but without much play, within the pile-case G, G. It is furnished with suitable recesses for the securing of the hammer, piston-rod, &c., and for enabling it to rise clear of the cylinder stuffing-box; and at its upper extremity a recess in the form of a species of inclined plane o', o'', is provided, for the purpose of acting upon the valve lever so as to permit the escape of the steam, after it has raised the hammer to a sufficient height.

P, the hammer: a cylindrical block of cast-iron, formed with a slightly concave face, fitted into the hammer-block, and fastened thereto by a wrought-iron key *o*, which at the same time serves to secure the connection of the piston rod.

Q, the latch-lever working in a recess in the hammer-block, (see fig. 16). The action of this part of the apparatus is fully described in the account of the steam hammer.

p, a small solid piston working in a cylindrical part of the valve chest, and attached to the valve by a short connecting rod. Its under surface is constantly acted upon by the pressure of the steam within the valve chest, so as to cause the steam-valve to assume the position indicated in fig. 9, unless counteracted by a superior force.

q, the valve spindle, produced downwards and working in suitable bearings, so as to bring it under the action of the trigger at the termination of the stroke.

r, the valve lever, working outside of the pile-case, but having a small friction roller attached to its inner end, and situated so as to come under the action of the inclined plane *o'*, *o''*.

s, the trigger, the function of which is to keep the steam valve in such a position as to prevent the admission of steam into the cylinder during the descent of the hammer-block.

t, the parallel bar, against which the latch lever acts at the termination of the stroke, for the purpose of releasing the valve spindle from the trigger, in order to allow the steam to be admitted for a fresh stroke.

u, *u*, *u*, the parallel motion bell-cranks and connecting rod of the disengaging apparatus.

v, a buffer-box for the purpose of restricting the travel of the valve to its proper amount, and of deadening the shocks to which it is subjected.

The above enumeration, together with the explanations previously given, will convey a sufficiently clear idea of the construction of the pile-driving apparatus, properly so called. I shall now proceed to describe the mechanism by which the pile-case and its appendages are elevated to the top of the great vertical guide-pole, and the means employed to render the whole machine locomotive.

The power requisite for these purposes is supplied by a small horizontal steam engine R, situated opposite to the great upright C, and under the boiler I, from which it derives its supply of steam. The motion of this steam engine is transferred to the axis of the great chain barrel by means of a train of spur geering, calculated to increase the power to the required extent. Two broad plates of wrought-iron, extending across the entire platform, and bolted securely to its timbers, afford a sufficiently firm foundation for the engine, and for the bearings of the various shafts in the train of wheels, which consists of three pairs marked *w*, *x*, and *y*, the pinion of the first pair being fixed upon the crank-shaft of the engine, and the wheel of the last upon the axis of the chain barrel. The pinion *y*, is fitted to slide longitudinally upon its shaft by means of a sunk feather, and is commanded by the lever-handle *z*, so as to enable the attendant in charge of the machine to throw it out of geer with the wheel upon the chain barrel shaft, when the latter is to be left free to revolve during the driving of the pile. The wheel *x*, upon the third motion shaft, geers with a similar wheel *a'*, fixed upon a cross shaft T, working in bearings under the platform, and serving to impart motion at once to the small chain barrel *b'*, for hoisting the piles, and to the locomotive geer. A clutch, or coupling *c'*, sliding upon the shaft T, enables the attendant to throw the small chain barrel into geer with the driving apparatus, or disengage it at pleasure; the remaining details of this part of the process will be fully understood by reference to fig. 1, where a pile H' is shown suspended from the chain *d'*, ready to come under the action of the driving machinery. To adjust the pile-case over the head of the pile at the commencement of the driving, it is of course necessary that one or two men should be raised to the summit of the machine. A rope *e'*, passed over a pulley at the top of the great upright, and wound round the barrel of a winch U, serves to accomplish this object.

The locomotive geer is exceedingly simple, and will be at once understood by referring to figs. 3 and 4. A bevil wheel *f'*, fixed to the outer end of the shaft T, geers with another of equal diameter, working loose upon the shaft V, to which a pair of the locomotive wheels *a*, *a*, are fixed. When it is required to move the platform with its superincumbent machinery along the line of rails, a sliding clutch *g'*, is thrown into geer with the last mentioned bevil wheel, and is disengaged when the machine has arrived at the desired position.

Action of the Machine.—The pile having been raised by means of the hoisting apparatus, and its point having been set into the proper position, the pile-case G, G, with its attached machinery, is lowered down over the head, by reversing the small engine R, so that the jaws *f*, *f*, rest upon the shoulders of the pile, which sinks down into the ground by the effect of the superincumbent weight, till it has reached soil sufficiently firm to support it; this is indicated by the chain *c*, *c*, becoming slack. The pinion *y*, is then thrown out of geer, and the steam is admitted into the driving cylinder F by turning the handle *i*. The hammer-block O is by this means raised till the inclined plane *o'*, *o''*, coming in contact with the end of the valve lever *r*, causes the valve *k*, to assume the position represented in fig. 11. The steam which had served to raise the hammer is thus allowed to blow out into the air, and the hammer descends and discharges its momentum in the form of an energetic blow upon the head of the pile. During the descent of the hammer-block, the steam-valve is retained in its proper position by the action of the trigger *s*, but by the effect of the concussion upon the head of the pile, the valve spindle is released from contact with the trigger, and the steam-valve assumes the position indicated in fig. 9, in which circumstances the steam is allowed to act freely under the piston, for the purpose of again raising the hammer.

Such is the rapidity with which these various movements and functions of the driving apparatus are accomplished, that the machine may be easily made to perform 80 strokes per minute. Some idea may be formed of the vast efficiency of this system of driving piles, when it is stated that, in ordinary ground, piles of 14 inches square are driven at the rate of upwards of 10 feet per minute!

At the conclusion of the driving of each pile the action of the hammer is arrested by turning the handle *i*, which cuts off the supply of steam. The great chain barrel pinion is then thrown into geer with the wheel on the barrel shaft, and the small engine R is started; which rapidly raises the apparatus off the head of the pile to the top of the great guide-pole C. In the mean time, the locomotive action is applied, and the machine brought opposite the next pile, when the process just described is repeated.

JAMES NASMYTH, Bridgewater Foundry,
Patricroft, near Manchester, April, 1847.

PLATE CXXXIII.

FAIRBAIRN'S VALVES AND EXPANSION GEER, APPLIED TO A SIXTY HORSE CONDENSING ENGINE.

This Plate is devoted to the illustration of an important practical improvement which has recently taken place in the application of the expansive principle to the steam engine; and in introducing this subject to our readers, we shall take occasion to give a brief historical account of the origin and progress of the system of working steam expansively, and a statement of the advantages which have resulted from the application of this important property of steam.

Our readers will find, in another part of this Work,* a full exposition of the nature and principle of the expansive action of steam, and of its economic uses, which were first discovered and applied by James Watt, in the year 1776. The mechanism which he adopted, and which, in some special cases, has not been excelled by any subsequent contrivances, consisted of a set of adjustable tappets fitted to the *plug-frame*, or air-pump rod of the engine, and acting upon the *hand-geer*, which is a series of levers connected with two sets of conical valves for regulating the admission and escape of the steam at the top and bottom of the cylinder. This arrangement is still almost universally employed in pumping and blowing engines, and is in many respects superior to all others that have hitherto been proposed, especially in the rapidity with which the stroke of the valve is effected, and the great range for adjustment which it affords.

For many years after the steam engine came into general use, the hand-geer constituted the best and most effective mode of applying its action to the valves, for the purpose of giving a continuous and self-acting motion to the machine, even in cases where the expansive principle was not taken into account. Subsequently, in cases where a rotatory motion was produced, the same object was effected by means of oscillating or revolving tappets, fixed upon a horizontal shaft driven either continuously by wheel-work, or with a reciprocating motion by an eccentric; these tappets or cams conveying motion to the conical valves by means of vertical rods. This arrangement is also frequently adopted in engines made at the present day; but as the importance of the expansive principle is now almost universally recognized, it is usual in modern engines constructed after this plan, to combine both arrangements; employing the eccentric oscillating motion to work the steam and exhaust valves, and the revolving cams to actuate the expansion valves.

Still more recently the principle of the slide-valve, in combination with the eccentric motion, (both which inventions are due to the late Mr. Murdoch, of Messrs. Boulton and Watt, to whom many of Watt's inventions owe much of their practical value), came to be generally applied; and the superiority of the arrangement in point of simplicity, efficiency, and durability, were so obvious as to lead to its almost universal adoption for general purposes. By this system the expansive action of the steam may, to a certain extent, be taken advantage of,—but it is attended with the inconvenience of *wire-drawing* the steam; in other words, of shutting the port so slowly, and opening it so gradually, that a considerable time expires before the steam is admitted with its full force upon the piston. To obviate this objection a separate expansion, or *cut-off* valve, is very frequently used in combination with the slide valve, but actuated by independent, and in most cases adjustable, mechanism, so as to secure as far as possible the advantage of the expansive principle. Several examples of this arrangement are represented and described in this Work.† It is defective, however, inasmuch as the action of the cut-off valve takes place at too great a distance from the cylinder; and it is seldom of much value unless the engines and boilers are capable of bearing a pressure of 15 to 20 lbs. on the square inch.

Of all the contrivances which have, from time to time, been proposed for the purpose of neutralizing the effect of the steam in obstructing the action of the valves, none has been so effectual as the introduction of the Cornish double-beat, or equilibrium valve, the invention of Mr. Joseph Hornblower, and improved by Woolf. This description of valve, the details of which are fully described in other parts of this Work, is peculiarly applicable to the purpose of working steam expansively, as it combines in a high degree the essentially requisite qualifications of celerity of operation and facility of adjustment to the precise moment of time and exact amount of opening required. Mr. Fairbairn, to whom the merit of applying this admirable kind of valve to rotative engines is chiefly due, makes use, in the example which we are now about to introduce to the notice of our readers, of a modification of the double-beat valve somewhat different to that employed in the Cornish engine.

The advantage to be derived from the expansive action of steam may be stated, in general terms, to be directly proportional to the pressure at which the steam is in the first instance applied to the engine. The use of high pressure steam in combination with its expansive action and the power derived from its condensation, originated in Cornwall, where the high price of coal, and the consequent necessity for rigid economy, together with the system of registration and encouragement held out by premiums, have afforded the greatest scope for ingenuity in the improvement of the steam engine as applied to the draining of mines. Accordingly, we are indebted to the skill, acuteness and careful attention of the Cornish engineers for the demonstration of the great advantages attendant upon their system (which is found to be equally applicable to rotative engines for manufacturing purposes); and the excellent examples which they have set are beginning to be appreciated and followed in other parts of the kingdom, even where the same urgent necessity for economy does not exist.

Ten years ago the average or mean expenditure of coal per indicated horse power was generally computed at from 8 to 10 lbs. per hour; now it is under 5 lbs. in engines that are worked expansively, and even this rate is far below the duty of a well-constructed Cornish engine, which ranges from 2½ to 3 lbs. per horse power per hour.

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* See Practical Essay on the Steam Engine, page 16.

† See Plates LVI.—LXXVII., with Descriptions, and Practical Essay on the Steam Engine, page 22.

This difference in the consumption of coal may be attributed to two causes: First, the conditions under which the duty of the two distinct species of engines, (namely, the Cornish, and the rotative engine for manufacturing purposes), are respectively performed: the former, being chiefly employed in pumping water, has the benefit of alternate action in overcoming the inertia of a large mass of matter, which, when once in motion, is more easily continued in that state for a definite time than in cases where the resistance is constant, as in corn or cotton mills; Secondly, the greater care and attention which the Cornish engineer pays to his boilers, steam pipes, cylinder, &c. Such parts are never left exposed, but are carefully wrapped up in warm jackets, and well clothed, to prevent the radiation of heat, (see Practical Essay, page 49).

But one of the principal reasons for the practice of rigid economy in fuel is the reduction of profits upon articles manufactured by steam power. Under such circumstances, a saving in fuel becomes a consideration of the utmost importance; and these reductions alone may be regarded as having afforded the powerful stimulus towards improvement in that direction which has recently manifested itself. To this cause especially may be attributed the increasing employment of high-pressure steam, and the application of its expansive action in enhancing its mechanical effect in the steam engine. In France and most other parts of the Continent, this system has been long in use, and although its economical results have been repeatedly promulgated in this country, it is only within the last few years that its advantages appear to have been duly appreciated. Indeed, for a great number of years, a strong prejudice existed against the use of high-pressure steam, and it required more than ordinary care, caution, and circumspection to effect the changes which have been introduced. Mr. Fairbairn, of Manchester, was among the first in the manufacturing districts who pointed out the advantages to be derived from the use of high-pressure steam and the expansive principle, and for many years he had to contend with the fears and the prejudices of the manufacturers before the present system of economical working was generally adopted.

The description of engine mostly in use on the Continent, and which is found to work most admirably in respect of economy of fuel, is *Woolf's double-cylinder engine*, in which two steam cylinders are employed, their piston rods both acting upon the same limb of the beam; one of the cylinders receiving the steam at full pressure from the boiler, while into the other, which is of considerably greater capacity, the same steam is admitted to act by virtue of its expansive energy, and in combination with the power derived from condensation. This arrangement is no less judicious than ingenious, and is advocated by many whose opinions are entitled to the utmost respect; it involves, however, considerable complexity of the mechanism, and proportional liability to derangement. Further, it is capable of demonstration, that the total mechanical effect of a given quantity of steam in the double cylinder engine is not greater, either theoretically or practically, than in one of the ordinary construction, working at the same rate of expansion, and with a well regulated expansive apparatus. It therefore becomes a matter of the utmost importance to determine what is the most appropriate, simple and effective mechanism for the attainment of that object.

We have already described and briefly criticized some of the most ordinary appliances for working steam engines expansively. These are all more or less liable to the objections: First, of wire-drawing the steam; Secondly, of cutting it off too slowly; and Thirdly, of leaving too much space between the cut-off valve and the cylinder, whereby much steam is wasted without producing any effect. To remedy these defects, Mr. Fairbairn employs an improved arrangement of valves and expansion apparatus, which is applicable to all rotative steam engines working expansively, whether with high or low pressure steam. This arrangement it is the object of the Plate now under notice to exhibit and illustrate.

Fig. 1 represents an external front elevation of the nozzles, side pipes, and cylinder of a sixty horse condensing engine, upon the new and improved system.

Fig. 2 is a side view of the same.

Fig. 3 is a transverse section of the side-pipes, steam chests and valves, shewing the general arrangement, and the progress of the steam from the steam-pipe to the condenser.

It will at once be observed by all who are conversant with the construction of steam engines, that the relative positions of the steam cylinder A, the cylinder bottom B, the upper and lower valve chests C and D, the steam-pipe E, and side pipes F and G, are identical with those of all engines of this description. The internal construction, however, of the steam chests and valves, and the mode of working them, are peculiar, and constitute the distinguishing feature of this invention.

It is of the utmost importance, in any arrangement of this nature, that the valves for the admission and escape of the steam should be situated as near as possible to the ports leading into the cylinder. Accordingly, in the figures now before us, it will be observed that the valve chests C and D, and the steam and exhaust valves *c*, *d*, *e*, and *f*, are placed immediately over against the upper and lower ports *a*, and *b*, respectively. The upper valve chest C receives the steam from the boiler by the steam pipe E, guarded by the throttle valve *q*, and stop valve *r*; while the lower chest D, connected to the upper by the side pipes F and G, is brought into communication with the condenser by the exhaust pipe H. The valves *c*, and *d*, next to the steam pipe E, are those by which the steam is admitted into the cylinder; and those marked *e*, and *f*, are the exhaust valves, by which the steam escapes from the cylinder to the condenser. All the four valves are of the same area and dimensions, but when in action the steam valves are not raised so high as the exhaust valves, for reasons which we shall presently specify. The arrows indicate the course of the steam in its passage from the boiler into the cylinder, and thence into the condenser; it will be observed that in the position of the valves, as shown in the figures, the piston would be in the act of ascending.

The construction of double-beat valves here employed by Mr. Fairbairn, is somewhat different to that of the ordinary equilibrium valve, (as represented in Plate LXXVII.,*) the present arrangement being more conducive to that compactness and proximity to the steam ports which we have already stated to be so desirable an object. They consist each of two single conical valves very carefully secured together, and accurately fitted to their seats; the *downward* pressure of the steam upon the top of the upper valve being in a great measure neutralized by its *upward* pressure upon the lower surface of the under, to which it has free access, as shown in the figures. The upper valve is made greater in area than the lower, in the proportion of 1·158 : 1, for the purpose of giving such slight preponderance to the downward pressure as to keep the valve always firmly pressed upon its seat, except when raised by the working cams.

The mode of working these valves is remarkably simple and efficient. Two horizontal discs *g*, *h*, carrying each upon its upper surface a series of cams or projections of definite length and height, are fixed to a light vertical shaft *i*, *i*, passing continuously through the centre of both valve chests C and D, and working in bearings formed in the upper ends of two hollow tubes cast in the valve chests for its reception; the lower end of the shaft is supported in a foot-step fitted into the top of an arched bracket bolted to the foundation. The shaft derives its motion, by means of a pair of meter wheels *k*, *k*, from a horizontal shaft *l*, driven by another pair of meter wheels from the fly-wheel shaft of the engine. Thus the cam-rings *g*, and *h*, revolve between the valve spindles, uniformly in the same direction, and with a velocity equal to that of the crank-shaft.

The rotation of the cams produces, by a direct and simple action, the requisite motion of the valves; the height to which they are raised, or in other words the area of aperture, being determined by the height of the cam above the plane of the disc, and the time during which they are retained open being regulated by the length of the face, which latter is made variable for the steam valve. The manner in which the action of the cams is transferred to the valves is as follows:—To each of the valve spindles, which work through stuffing-boxes on the valve-chest covers in the usual manner, a small cross-head *m*, is firmly attached by adjusting nuts, and is guided at its outer end by a fixed vertical rod *o*, and at the opposite extremity by a brass piece *p*, also fixed to the top of the rods *o*, *o*, and having vertical slots formed in it for the reception of the ends of the cross-heads; the pieces *p*, *p*, are furnished also with sockets for the guidance of the upper ends of the valve spindles. On each of the cross-heads *m*, *m*, is mounted a small friction pulley or roller *n*, adapted to bear against the surfaces of the cams; in the case of the steam valves, these rollers are capable of being adjusted, by means of a small apparatus shown in the figures, so as to bring them over any section of cam that may be required; the small indices serving to point out to the engineer the exact amount of expansion at which the engine is working.

By the above arrangements it will be perceived that the valves will be raised and retained open or shut for any definite period of time; the preponderance of downward pressure which we have already explained as due to the peculiar construction of the valves, being just sufficient to act as a *spring* to keep the friction rollers constantly pressed against the surfaces of the cams, and to overcome the friction of the packing in the stuffing boxes, &c.

It has been already stated that the steam valves are not raised so high as the exhaust valves, and for this obvious reason, that the exhaust being a constant quantity and requiring full openings into the condenser, it is desirable to retain them open throughout the whole length of the stroke. By the present arrangement this is effected with a greater degree of certainty and precision than by any other hitherto proposed: the exhaust valves rise suddenly upon the short inclined planes of the cams, and, having allowed time for the escape of the steam from the cylinder through a wide passage into the condenser, they fall with equal celerity by their own weight; and thus a more complete vacuum is formed under the piston than it is perhaps possible to obtain by any other process.

The *stop*, or as it is sometimes called, *governor valve r*, is a simple conical valve, guided beneath by a stalk working through a socket in its seat, and above by a spindle passing through a stuffing-box. It is commanded by means of a lever *s*, and rod *t*, the lower end of which is screwed and fitted into the eye of a hand-wheel *u*, revolving on a bracket or arm *v*, so as to bring the valve within the control of the engineer.

Jan 6
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Rush

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